MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Machine-Shop Practice

The "Master Tools of Industry," as machine tools have been designated, are basic implements, as all kinds of accurate machine construction depends absolutely upon them. The A.S.M.E. through its Machine-Shop Practice Division is devoting increasing attention to the design and economical operation of these fundamental instruments of modern mass production. In this issue are three papers on important phases of machine-shop practice which are to be presented under the auspices of the Division at the technical sessions in connection with the Machine Tool Exhibition in New Haven on September 8–11, 1925.

SEPTEMBER 1925

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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ADVERTISING

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Coming A.S.M.E. Meetings

Machine-Tool Convention, New Haven, Conn.
Sept. 8–11, 1925

Regional Meeting, Altoona, Pa.
October 5–7, 1925

Annual Meeting, New York, N. Y.
Nov. 30–Dec. 3, 1925

MECHANICAL ENGINEERING

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Theory and Practice of Centerless Grinding

Advantages and Disadvantages of Centerless Grinding—Dressing for Through-Grinding—Work Guides—Producing Round Work on the Centerless Grinder—Production by the Through-Grinding Method—Straight-in or Form Grinding

By W. J. Peets,[1] Elizabethport, N. J.

THIS paper is a treatise on the modern theory and practice of centerless grinding as carried on at the present time—with photographs showing various actual examples of special set-ups for a variety of work and production data secured from actual practice. The author does not wish to give the impression that the matter presented is the result of his own research work, for it is but a compilation of present-day knowledge of the art, combined with the results of practical experience with this kind of machinery in a manufacturing plant.

The centerless grinder has come to play an important part in the quantity production of cylindrical ground work, and on work to which it is adapted this type of machine outclasses any other for economy in production.

First, let us study the elements of the centerless grinder and learn what advantages and disadvantages it has when compared with the between-centers grinder.

The centerless grinder consists primarily of two abrasive wheels mounted so that their peripheries face each other, one of the wheels having its axis so arranged that it can be swung out of parallel with the axis of the other wheel by varying amounts, as required. Between these two abrasive wheels is a rest which supports the work. (Fig. 2.)

The two wheels are run at different speeds, the fast-running or grinding wheel at regular grinding speed, say, 5500 to 6000 ft. per min. at the periphery, and the slow-running or regulating wheel at the work speed only, which latter varies according to the material, length, diameter, etc. of the work being ground. The wheels are revolved so that their adjacent faces move in opposite directions; therefore a piece of cylindrical work placed on the rest between the wheels and touching both of them will be revolved by contact with their two surfaces. The wheels are revolved at different peripheral speeds because, if both ran at one speed, the work placed between them would be merely revolved as an idler gear, with no grinding action. The difference in surface speed of the two wheels imparts the necessary relative motion for grinding.

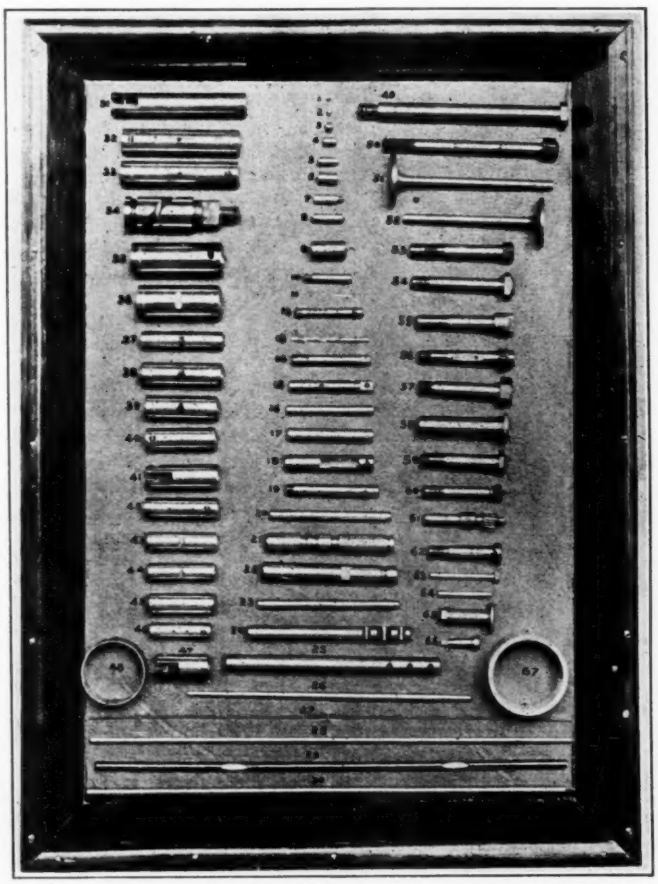

Fig. 1 Centerless-Ground Pieces

The success of centerless grinding lies fundamentally in the fact that the friction of rest is greater than that of motion. For this reason the work between the two wheels will take the same peripheral speed as the slower-running, or regulating, wheel, as if geared to it. Thus we have the cylinder to be ground revolving between two abrasive wheels of different surface speeds, but turning as if geared to the slower of the two, and being held against the work rest by the pressure of the grinding wheel.

Now, if the axes of the two wheels remain parallel, the cylinder placed between them will receive no motion except that of rotation. However, if the axis of one wheel is thrown out of parallel with that of the other, as well as out of line with the work rest, a lengthwise motion will be imparted to the work, causing it to travel forward past the wheel surfaces by a certain amount per revolution of the regulating wheel, according to the adjustment given that wheel for this purpose. (Fig. 3.) Thus the speed of passage of the work through the machine is governed by changing the angle α between the axes of the two wheels.

Therefore, according to the relative axial position of the wheels, the centerless grinder may be employed for two distinct classes of cylindrical grinding. (1) By setting the axis of the regulating wheel parallel to that of the grinding wheel and feeding the regulating wheel straight in toward the grinding wheel, the work may be ground without lengthwise travel. This affords wide application of centerless grinding to all sorts of shouldered work, such as studs, etc. having one or more diameters, and is known as "straight-in grinding." (2) By setting the axis of the regulating wheel at an angle to the axis of the grinding wheel as shown in Fig. 3, the work is kept in constant lengthwise motion through the machine, this being known as "through-grinding." In this latter case there is no forward movement of the regulating wheel, except as it must be advanced by the operator from time to time to compensate for wheel wear, or its distance from the grinding wheel changed to accommodate different sizes of work.

Advantages of Centerless Grinding

1 The chief advantage of centerless grinding over between-centers grinding is that the action (in through-grinding) is con-

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tinuous. This means that the machine is grinding all the time, and that the time waste which occurs in between-centers grinding incident to placing the work between centers, putting on and removing dogs, advancing the wheel to the work, etc. is eliminated. In fact, on work of small or medium diameter and length the operator is kept very busy simply feeding the machine.

2 A much smaller stock allowance for grinding is necessary than in between-centers grinding due to all errors in centering being eliminated and to the fact that only enough stock need be removed to take away the tool marks. While it is necessary to remove from 0.006 in. to 0.015 in. when grinding between centers, as little as 0.00075 in. is sometimes sufficient in centerless grinding.

3 The work is as a rule supported better in centerless grinding. This is especially true of pieces which are long in proportion to their diameter.

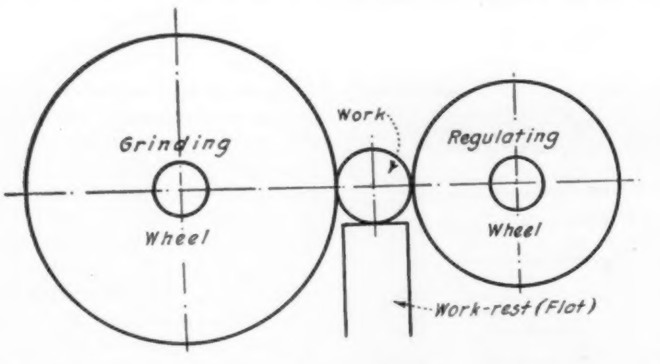


Fig. 2 Centerless Grinding—Work on Rest between Two Abrasive Wheels

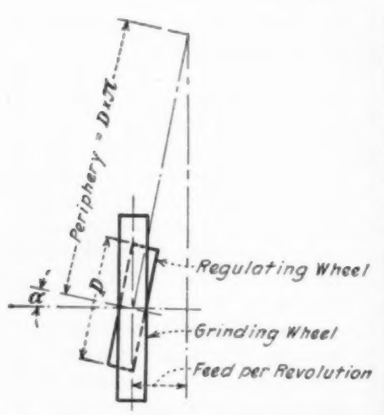


Fig. 3 Adjustment of Regulating Wheel for Feeding

(Feed of work in inches per minute = $\pi DN \sin \alpha$, where N = r.p.m. of regulating wheel.)

4 The size of work can be held to closer limits on the centerless grinder with less skill on the operator's part, as the wheels are not moved during grinding, and the chances of error arising from infeed of slides, stops, etc. are eliminated. Also, in centerless grinding stock is removed from the diameter instead of from the radius of the work, which reduces by one-half any error due to wheel wear (as compared to between-centers grinding), while the wheel wear is minimized by the comparatively light cut taken, making unnecessary frequent adjustments for holding size of work.

5 This simpler type of machine, besides requiring a less skilled operator, also effects a considerable reduction in upkeep expense, as the only moving parts while operating are the two wheel spindles and the mechanism for turning them.

6 No centering is required.

Disadvantages of Centerless Grinding

1 The main disadvantage, or limitation, of the centerless grinder is that quantities of work must be fairly large to warrant setting up the machine.

2 The type of work is limited to pieces of one diameter for through-grinding, and of one or two diameters for shoulder work in straight-in grinding.

3 The centerless grinder cannot hold the outside of a bushing concentric with the hole, and for best results, flats and keyways or oil channels (especially those running to the end of the piece) should be avoided.

4 Exact roundness such as is required for gages, etc. is harder to obtain than on the best types of between-centers grinders. By this is meant uniform roundness well within a tenth of a thousandth of an inch (0.0001 in.).

Thus the field of usefulness of the centerless grinder may seem somewhat restricted, but there is an ever-growing variety of work which can be ground by this method, or which can be designed so as to be so ground.

We may say that this type of machine is preëminently suited to straight cylindrical work of one diameter, with no exact reference to bore, and with no prolonged breaks in the surface.

Practice in producing straight centerless-ground work varies considerably. In some shops considerable metal is left on for grinding—in some cases as much as 0.016 in. to 0.020 in., and even $^1/_{32}$ in. Where so much is allowed for grinding, more passes through the machine, and at a slower speed, are necessary. This means greater labor cost, as well as greater wheel wear and cost. As a rule, as little stock as possible should be left for removal by centerless grinding.

In many cases the limiting speed factor is the ability of the operator to feed the machine, rather than the capacity of the machine itself. The peripheral speed and the grade of wheels used are the same as for the same work on any other grinder. However, by leaving a minimum of stock to be removed, a finer-grain wheel may be used and a finer finish obtained. For through-grinding, however, the grinding wheel should be soft enough to dress itself, or wear down by contact with the work, and not glaze or load up. When a wheel is soft enough to dress itself in this way, frequent diamond dressing is unnecessary, and it may be necessary to dress only once in several weeks by passing an emery stick across the wheel face to remove small particles of metal. For this reason wheels made with a rubber or an elastic bond are often successfully used.

Dressing for Through-Grinding

For through-ground work the dressing of the regulating wheel is really more important than that of the grinding wheel. As the axis of the regulating wheel lies at an angle to the work, it is necessary that the dressing of this wheel should occur at the same angle. One way of accomplishing this would be to remove the work rest from the machine and put in its place a dressing slide moving in the same plane and carrying the diamond at the same height in relation to the wheel center as the point of contact between the work and the regulating wheel. When the diamond traverses the wheel in this position, the result will be a perfect contact of the regulating wheel with the work during grinding—which is the requirement. Again, the dressing slide for the regulating wheel may be mounted at some other point around its periphery, and correct results obtained by swiveling this slide to the same angle to which the wheel is swiveled and moving the slide or diamond tangentially above or below the wheel center to correspond to the contact of the work with the wheel.

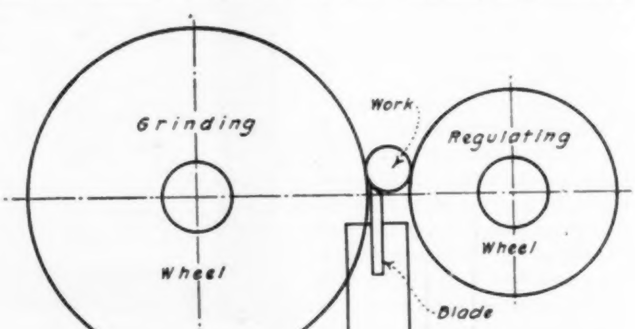


Fig. 4 Arrangement for Rounding Up Out-of-Round Work

The grinding wheel is dressed parallel to the regulating wheel by means of a suitable slide. However, as soon as work is introduced between the wheels, the grinding wheel gradually dresses itself away at the entering side until cutting takes place all across its face, the final sizing being accomplished as the work leaves the wheels. It will thus be seen that the original parallel dressing of the grinding wheel is lost after a few pieces have passed through and that instead of the faces of the two wheels remaining parallel,

they are out of parallel by the amount of stock which is removed during one pass through.

Here will be seen a further necessity that the grinding wheel be soft enough to dress itself by contact with the work passing through. The regulating wheel has practically no grinding action and so does not wear away as the work is rolled upon its surface. It is common practice to have the regulating wheel somewhat harder than the grinding wheel.

Work Guides

It is necessary in through-grinding to have guides of some sort to start the work in line with the wheels and to guide it as it leaves them. These guides are usually parallel strips of hardened steel, adjustably mounted so as to form side boards along that part of the work rest which projects beyond the wheels. Facility of adjustment is an important factor in setting these guide plates, and some experience is necessary to do it properly. Generally speaking, the guides should extend at the front and back of the machine to a distance equal to the length of the work being ground, as they are particularly necessary on long work of small diameter, which has a tendency to whip if not properly supported. It is not always possible to extend the guide plates this far from the machine, especially when grinding shafting, drill rod, etc. and external guides are usually provided when such long pieces are ground.

When setting the guide plates, care must be taken not to cramp the work as it enters or leaves the wheels, as this will result in tapered work; on the other hand, guide plates too loosely set may cause chatter on long work.

Sometimes a desired taper is produced by purposely setting the guide plates so as to cramp the work as it leaves the wheels. This method is not recommended where great accuracy is necessary.

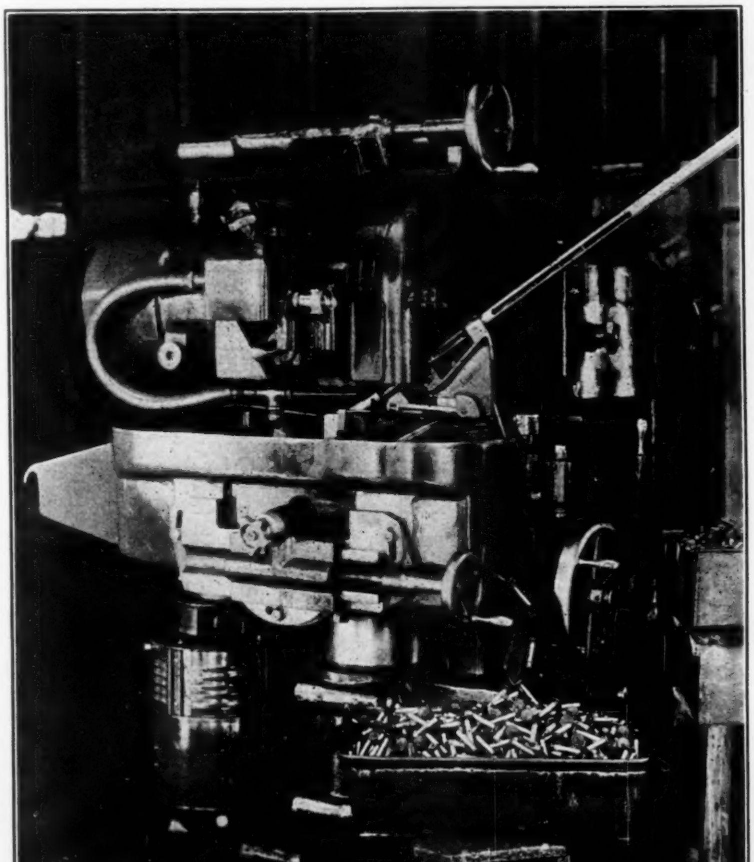


Fig. 5 Magazine Feed for Studs

Can Round Work Be Produced by the Centerless Grinder?

A common question regarding centerless grinders is, "Can round work be produced from stock which is originally not round?" This is a natural question for two reasons. First, the work while being ground is not held on any center but is free between the wheels. Secondly, out-of-round work will be produced by centerless grinders under certain conditions.

Experience with such out-of-round work has led many persons to state that the centerless grinder is only a sizing machine and that it cannot generate round work. The fact that out-of-round work may be produced on this machine under certain conditions makes it wise to have some means for testing roundness other than a snap gage or micrometer. A good method is to use a V-block, over which is mounted a dial gage or other registering instrument. A piece of work revolved in a gaging device of this kind readily registers any out-of-roundness which may be present.

With the proper setting, round work can readily be produced by the centerless grinder. By this is meant roundness registering to a limit of $^1/_{10}$ of a thousandth (0.0001 in.) in the V-block test.

For a machine to round up work originally out of round, the most important thing to consider is the relation of the center of the work to the centers of the two wheels. If a piece of work elliptical in shape, or having flats in its diameter, is placed in the machine so that its center is directly in line with the centers of the two wheels while resting on a flat work rest, an out-of-round product will result. The out-of-round piece will usually tend toward a three-cornered shape. It will gage correctly for size at any point all around, but if given the V-block test it will be found to be far from a true cylinder.

In general, the reason for this is that, as the regulating wheel is directly opposite the grinding wheel, a depression at any one point in the work's surface will produce a high point on the side of the work directly opposite—the approximate center of the piece simply moving back and forth across the flat work rest. (Fig. 2.) Once this three-cornered condition is established, further passes through the machine will do little or nothing to remove it. This action has probably not been mathematically analyzed, but observation indicates that the above is the case.

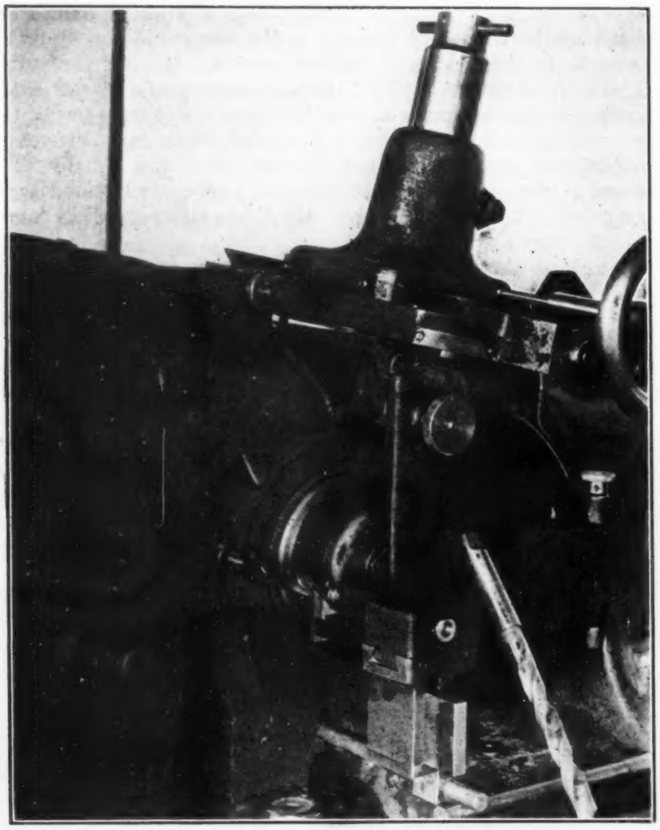
Now, if the work is elevated so that its center is somewhat above the center line of the wheels and a 45-deg.-angle rest is substituted for the flat one (Fig. 4), a condition is reached which tends to round up the work, even though it is presented to the machine in an out-of-round form. This is partly due to the fact that as the work is now above the center line of the wheels, the points of contact of the wheels upon its surface are not directly opposite each other as regards the center of the work, so that a high point on the work will not now produce a low point directly opposite. Also, the contact of this style of work rest with the work is very near the grinding wheel, and not directly below the work. Here again probably no mathematical analysis has been evolved to explain the action which takes place. However, the fact remains that commercial cold-drawn or even hot-rolled steel, by proper care in setting up and by sufficient passes through the grinder, will be finished round within the aforementioned limit of 0.0001 in., as well as accurate to size within about 0.0002 in., or perhaps closer. It is therefore evident that the setting of the work rest is a very important matter where accurate round work is necessary.

Work rests are subject to considerable wear, and are usually made of hardened steel or stellite. The latter material has been found superior, being usually brazed into a steel plate and ground to shape.

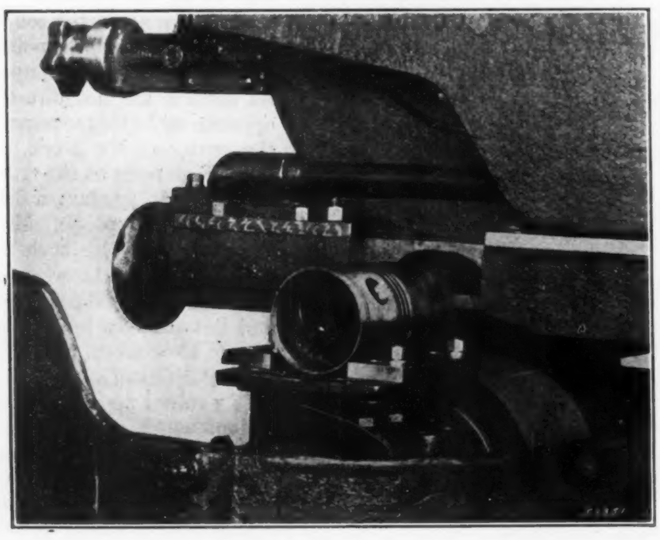
The condition of the cooling compound used is also a factor in the wear of the work rest, and the cleaner this compound is kept, the less wear will take place. This necessitates an ample settling tank with a proper strainer to remove the particles of metal and abrasive which will otherwise be poured over the work and work rest.

Production by the Through-Grinding Method

The rate of production of work by the through-grinding method



Grinding Taper Shank on Twist Drills



Finish-Grinding Pistons

will vary greatly according to the material and diameter of the work, the amount of stock to be removed, and the accuracy of finish wanted. Where the work is comparatively large and long, the output depends entirely upon the capacity of the grinding wheel and the power of the machine, while on short pieces of small diameter, where little stock has been left for grinding, the capacity of the machine may exceed the ability of the operator to load it.

To quote from actual experience, hardened shafts $^1/_4$ in. to $^3/_8$ in. in diameter and from 4 in. to 16 in. long may be ground round, within limits of $^1/_2$ thousandth (0.0005 in.) in diameter—removing from $^3/_4$ to 1 thousandth of an inch in one pass through the machine, at the rate of 12 to 15 ft. per min.—using a 100-grain elastic-bond wheel and producing a fine commercial finish. These same shafts may then be reground to a total limit of two-tenths of a thousandth (0.0001 in. $\pm$), by giving them an additional pass through the machine at the same rate. Where coarser wheels are used, this production can doubtless be bettered. Even at the rate of 15 ft. per min., forty-five 4-in. shafts are produced per minute (2700 per hour), receiving one pass through the grinder.

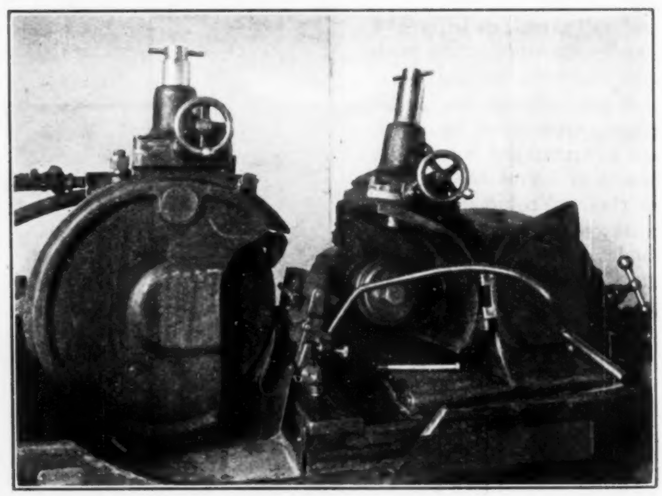
Many cylindrical parts having burrs thrown up by cutting-off, cross-drilling, or milling may be very economically burred by passing them through the centerless grinder, although they may not need grinding. It often pays to redesign certain parts so that they can be finished on the centerless grinder, because of its great capacity, accuracy, and low cost.

Where work is required to be concentric with a hole through it or with a smaller diameter, it is often the custom to rough-grind it between centers to a wide limit, and then finish-grind it in the centerless-type machine, thus securing the accurate finish of the latter at a combined cost which is often less than that of finishing the work between centers.

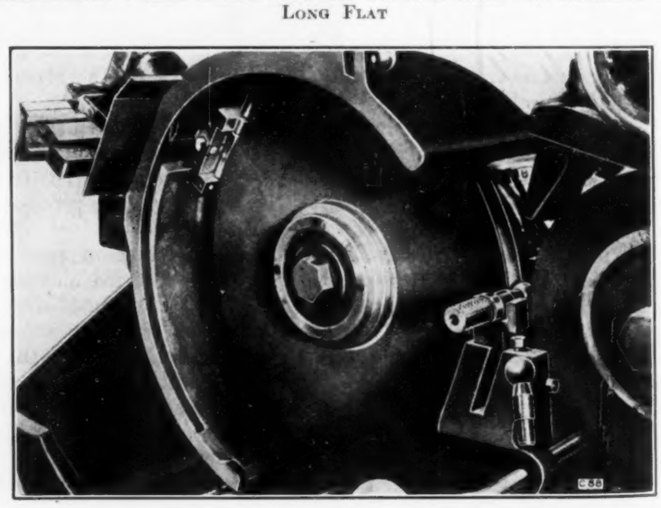
Straight-in or Form Grinding

This class of work is possibly more interesting than through-grinding because of its almost endless variety. The same machine is used as for through-grinding, with means provided for moving the regulating wheel toward and away from the grinding wheel with modifications, and with a means provided for moving the regulating wheel toward and away from the grinding wheel. This movement is accomplished by hand lever or power, as the case may be. The regulating wheel, as previously mentioned, is set at 0 deg. or as nearly so as possible, and the outside plates for guiding the work to and from the wheels are removed, only the work rest between the wheels remaining. The action is usually as follows:

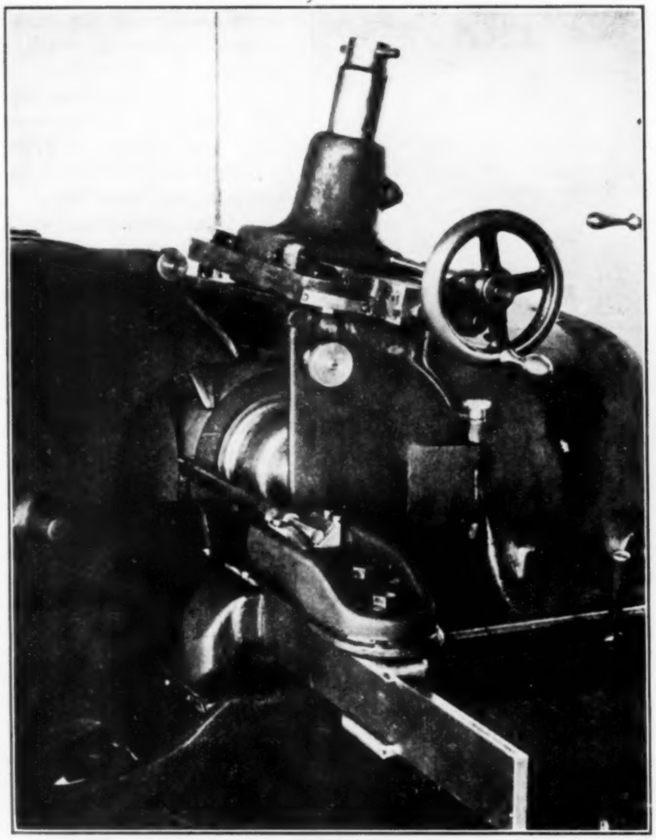
The regulating wheel is sufficiently withdrawn from the grinding wheel to permit the work to be placed between the wheels, either



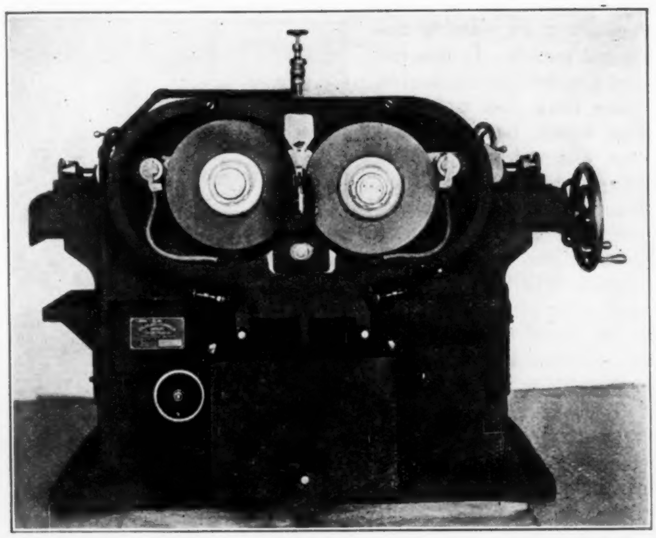
Regulating Wheel Dismounted and Fixture Used for Grinding a Long Flat



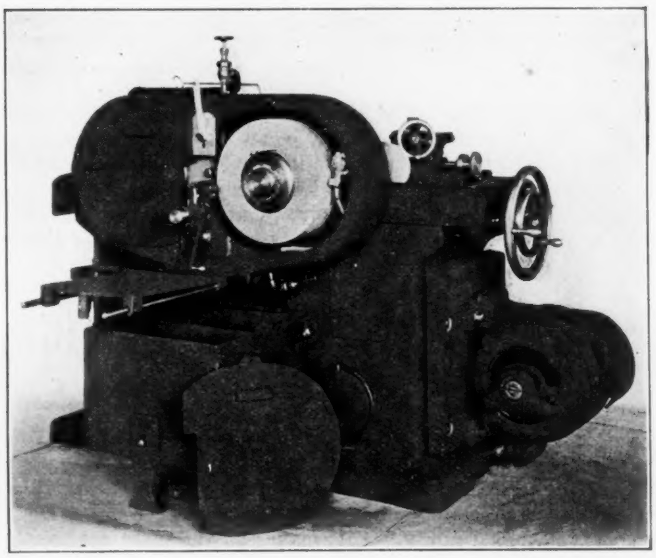
Grinding Ball Shape with Form Wheel



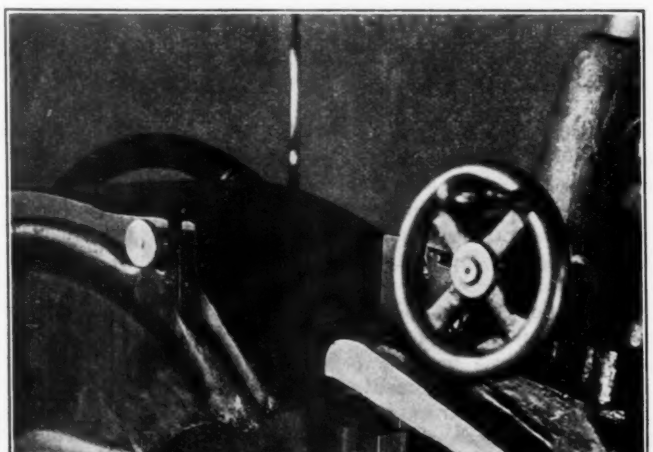
Straight-in Grinding on Ends of Long Piece



Set-Up for Through-Grinding



Machine Set Up for Spot Grinding Showing Elevating Work-Rest

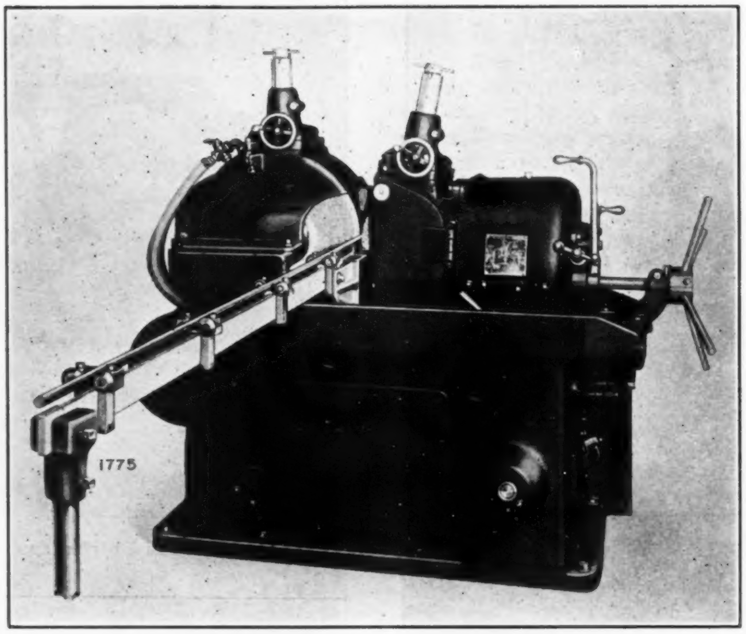


Magazine Feed for Grinding Rollers

Automatic Attachment for Straight-in Grinding

manually or by suitable mechanical means. In this position the work is supported by the work rest and regulating wheel, but does not touch the grinding wheel. The regulating wheel is then advanced, revolving the work upon the inclined surface of the work rest until contact is made with the grinding wheel, after which the grinding is done by means of further forward movement of the regulating wheel. After grinding, the regulating wheel is withdrawn and the work ejected or allowed to roll down between the work rest and regulating wheel and thus out.

Where large quantities of work are to be handled, hoppers and magazines are specially built therefor. Small work with a minimum of stock to be removed is thus ground in as little as three seconds per piece, including loading.



Roller Guide for Very Long Work

A great variety of work is today being ground by this method, including shouldered work such as automobile shackle bolts, tapered work, multi-diameter work, formed work, etc.

As wheels 8 in. wide and possibly wider are being used, and as dressing slides are being developed capable of producing tapers, forms, and radii, an almost endless variety of work is possible.

Roundness, size, finish, etc. are affected by the same elements in straight-in form grinding as in through-grinding, although the action is somewhat simpler due to the regulating wheel not being set at an angle. The angle of the work rest and the position of the work above the center line of the wheels are important in obtaining roundness on straight-in ground work for the same reasons as those applying to through-ground work.

In conclusion, it may be said that, as in the case of between-centers grinding, a great deal depends upon spindle construction and maintenance. If close limits are to be held (say, 1 or 2 tenths of a thousandth of an inch), both the grinding-wheel and regulating-wheel spindles must be run very close to their respective boxes, as a small displacement of either will result in loss of size and tapered work. This is particularly true of through-ground work, where full contact between the work and the wheels is not constant, as when single pieces are entering or leaving the wheels.

Coal Handling Below Ground

By NIXON W. ELMER,[1] QUINCY, MASS.

For many years it might have been safely said that the bituminous-coal industry led the world in conservatism and close adherence to precedent, and for many years progress underground has been mainly in the direction of larger units, better control, and added safety. This makes all the more noteworthy the sudden change that appears in the diversified series of experimental attempts to substitute mechanical means for hand labor underground.

As was to be expected, many of these efforts were fore-doomed to failure. For reasonable assurance of success in so radical a step several fundamental conditions must obtain:

(*a*) A thorough understanding is needed of all the essential conditions which must be met underground, in any particular mine

(*b*) A complete knowledge of the present state of the art of materials handling as developed above ground is necessary

(*c*) Whole-hearted backing and coöperation, based on a correct understanding of the time and money required for necessary experiment, adaptation, and application, is equally essential.

Taken together and correctly applied, these three conditions should prevent waste effort where insurmountable obstacles of a physical, financial, or mental nature exist.

Bearing in mind the fact that in these early experiments one or more of the above prerequisite conditions was usually absent, the progress which has already been made is astonishing and impressive. This in itself should cause us to anticipate rapid progress now that the problems are more clearly understood.

Almost every conceivable piece of handling equipment developed for some specific purpose above ground has been, at one place or another, hopefully introduced underground. The fact that even a few of these efforts have turned out profitably is encouraging.

The development has been along two general lines:

(*a*) One group has held tenaciously to every detail of present mining practice as developed through the years of cheap hand labor

(*b*) The other has unconsciously assumed that the conveying methods as developed above ground were fixed and therefore mining methods must be radically changed to fit.

Happily it is becoming generally understood that both these ideas may be incorrect, and encouraging progress is being made in the direction of designing and laying out suitable mechanical handling, while retaining the important features of present practice that affect safety. The difficulty here is to make no mistake in picking out these vital features from the mass of past practices, all equally venerated.

Obviously if any important safety features are changed disaster may close the experiment; on the other hand, if the designer is faced with an unnecessary number of fixed conditions, his design must work out correspondingly below par.

It is interesting to note that there is not a single operation between the firing of the shot at the face and the delivery to the common carrier of the product of the mine which is not now being done by conveyors of one type or another, at some mine, somewhere in the United States.

If we assume that the effort and therefore the progress will be in proportion to the reward, let us suppose a working condition where an ordinary man of average industry is able to load directly into mine cars at the face, ten tons per day. Then the same man with the same industry under ideal conditions, where the coal is carried away as fast as shovelled, where he never has to take a step with a full shovel, and where the maximum lift is twelve inches, can load about 30 tons.

Ideal conditions are never fully attained, but with a 33 per cent discount for this we still have a large enough saving for progress.

The effect which success in these efforts will have upon the coal-producing industry may be summarized as follows:

(*a*) An obvious saving in direct labor

(*b*) Reduced overhead, through greatly reduced workings

(*c*) A decrease in the number of companies operating.

[1] Consulting Materials-Handling Engineer. Mem. A.S.M.E.

Precise Cylindrical Lapping

By PAUL M. MUELLER,[1] HARTFORD, CONN.

THE cylinder and the plane are the two basic forms used in all machine design. They are the first two forms in that series of shapes any one of which, like the propositions of Euclid's Geometry, may be developed from those which precede.

Once we have planes and cylinders we can develop angular measure. Then comes the cone, and with this addition we can generate such complex surfaces as the helicoidal involute of a ground worm and the octoid of a bevel gear.

The production of precise cylindrical surfaces has always occupied the attention of engineers and mechanics. With the introduction of the cylindrical grinder came a precise and easy control, and quantity production to fine limits was possible.

However, the solution of the problem of making fine commercial cylinders exposed the problem of making still finer cylinders for gaging that production. In a great many cases these gaging surfaces must be better than can be produced by simple grinding, and the already familiar hand-lapping method is used successfully.

In Fig. 1 the manual method shown produces nice surfaces of a high finish, when carefully applied. It has two drawbacks in that it requires skilled workmanship and considerable time. In other words, it is expensive. Furthermore each unit of a lot of gages must be treated separately until finished. The operator cannot set the lap, go through a batch of gages and, without intermediate test, be sure that they will all be identical.

From a gagemaker's viewpoint these drawbacks are very irritating. The gagemaker wants to make a batch of gages, then inspect them and have them pass. Hand lapping and desirable routine are therefore in opposition.

For some time the Pratt & Whitney Co. have been successfully lapping flat surfaces by the Hoke method. The lapping machine is shown in Figs. 2 and 3. The emery-charged laps are stationary and the gages are pushed around between them by the cranked spider. The top lap floats and is constrained only against rotation.

As the work progresses, eventually all blocks are in uniform contact with each lap. If the laps are parallel each block is identical with every other block. To insure this parallelism a transposition is occasionally made in such fashion that the sum of the lengths of any two adjacent blocks is equal to the sum of the lengths of any other adjacent pair. Several transpositions are sufficient to remove all grinding errors, and a transposition each time the machine is stopped for measuring maintains the identity of size.

The machine stops automatically after a predetermined number of revolutions and the operator sets the revolution dial always to remove half of the remaining plus metal.

The success of the method is due to three features:

1 The rate of production is high
2 The rate of metal removal is low
3 Transposition eliminates size variation in a given batch.

It was obvious that some similar form of lapping for cylindrical work would have the best chance of success. Several trial set-ups of apparatus were made and one of them worked nicely; in fact, results were beyond expectation, for better surfaces were produced than could be made by hand lapping, and size control was so simple that the gage diameters could be kept just under the maximum value of the wear limit.

Method of Production

The first trials were made in the Hoke machine which has been described. Both laps were stationary as for flat work and the cylinders were in a slotted spider between the laps. The spider was given a cycloidal movement as shown in Fig. 4, so that a given cylinder went through the following phases in a half-cycle of the cranks:

a Pure sliding on diametral elements
b Combined sliding and rolling
c Pure rolling
d Combined sliding and rolling
e Pure sliding, and so on.

The sliding was to effect the reduction and the rolling to distribute the reduction uniformly. The theory was good but the work came out staved like a barrel. All the pieces were identical, however, and the remedy for the staving in sight.

Fig. 5 shows the apparatus which solved the problem nicely. The upper lap in this case rotates, but otherwise floats and aligns itself on the cylinders in the spider between the laps. The bottom lap is stationary as before. Thus in effect the assembly is a roller thrust bearing. It differs from a thrust bearing in that the cylinders are set at an angle to the spider radius and that the spider is off

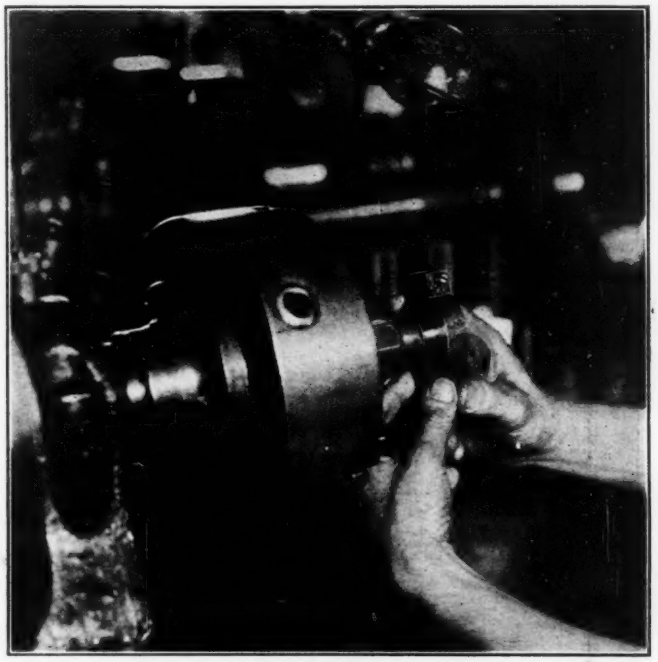


Fig. 1 Hand Lapping a Cylinder

center with the laps. The spider is free to rotate around a crankpin, and the crankshaft is slowly and positively driven from the upper lap spindle. The eccentricity of the crank is used to distribute the wear on the laps. The inclination of the roll axis to the spider radius is apparent.

Those who have used a planimeter know the action of the metering wheel with its combined sliding and rolling. Its analogy to the rolls in the spider is exact.

This last-mentioned method produces cylinders with high finish which are diametrically identical, parallel, and with straight elements. The precision is almost mathematical, but the work is not cylindrical. The cross-section is bounded by a peculiar lobed curve always having an odd number of lobes. The removal of these lobes will be discussed more fully later.

Rough- and finish-lapping operations are done on the same laps and equipment. The only distinction is that during roughing, free emery and kerosene are applied, and during finishing, cream rouge and kerosene are used with no free abrasive.

The roughing emery is usually SF 10X Washington Mills. This is a floated emery whose grain size is easily resolved under a microscope with a magnifying power of 25.

Preparation before Lapping

In general, work is prepared for lapping by grinding. For gage work we find it most economical when the grinding size is 0.0002 in. to 0.0003 in. above finish size.

When the quantity is sufficient, cylinders $^1/_2$ in. in diameter and

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Contributed by the A.S.M.E. Machine Shop Practice Division for presentation at the New Haven Machine Tool Exhibition, Mason Laboratory, Yale University, New Haven, Conn., September 8 to 11, 1925.

smaller are lapped from the hardening fire without intermediate grinding. Three-sixteenths to half-inch stock is rolled down a plane into the quench to keep it as straight as possible. Smaller diameters are hardened in the bar, broken up, and the ends of the short pieces smoothed preparatory to lapping.

When extremely hard surfaces are required, finish-ground cylinders are quenched from a cyanide bath. The thin scale formed is removed under the laps and the cylinders finished to size directly.

Preparation of Laps

To get the best work the laps must be as flat as it is possible to make them. They must be true abrasive planes. Fortunately this is not difficult to accomplish. They are divided into groups of three or four, and separate pairs of a group are rubbed together with fine emery in between them until they fit in any position. When this condition is reached the surfaces of that pair are either planes or spheres. After several pairs are fitted new combinations (concave with concave, etc.) are made and rubbed until they, too, fit. This procedure is carried on until any pair of laps selected at random will fit. When this has been accomplished we can say definitely that the surfaces are planes. Since if one lap is the complement of two others, these must be identical; and if, in addition, they fit each other we cannot imagine them of any other form than flat.

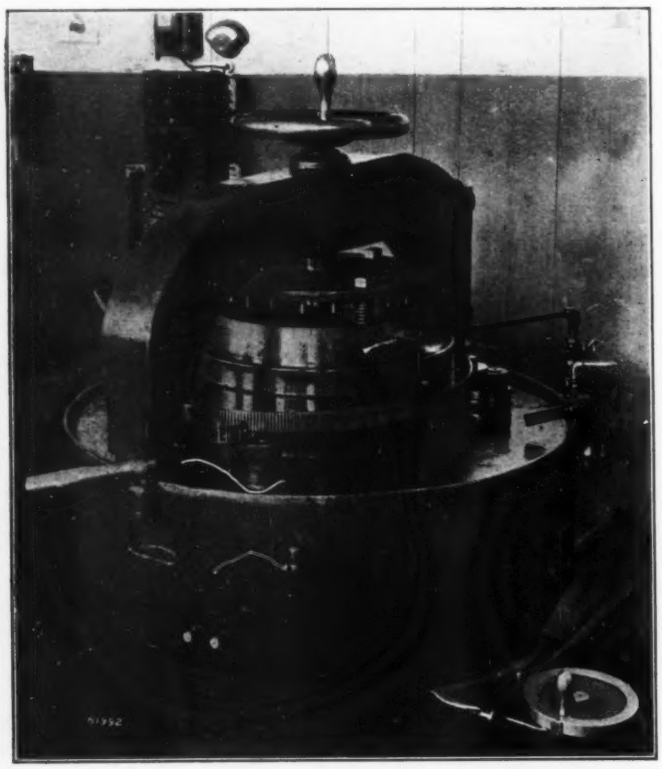


Fig. 2 Hoke Lapping Machine Closed

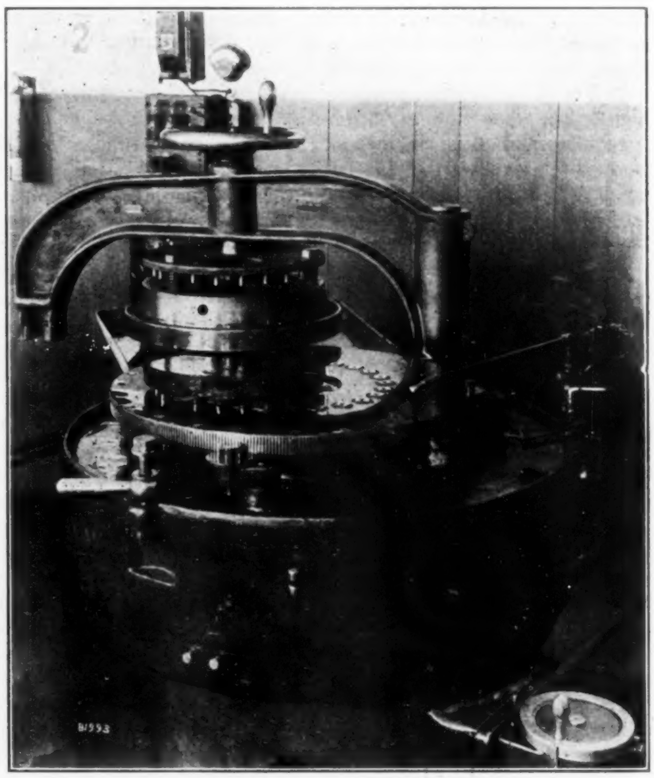


Fig. 3 Hoke Lapping Machine Open

A freshly dressed lap plane is a beautiful thing and looks to be anything but cast iron. After so much rubbing the emery is driven into the soft metal so uniformly that the surface is like the facet of a jewel. The face presents millions of tiny cutting edges so small that the scratches produced on the work are almost invisible.

It is a remarkable fact that the laps tend to improve with use. As the work slides and rolls about on the faces the tiny cutting grains are still further reduced in size by crushing, so that eventually they will not cut at all. It is due to this fact that free emery and oil are used while rough-lapping, in order to replace those grains which are continually being destroyed. The finish-lapping is begun when the work is of such diameter that the reduction to finish size will be complete when the emery has just "gone dead." When the emery is "dead," of course, there is a mirror finish on the work.

Lobing

Discussing in more detail the lobing of work which was mentioned earlier, the simplest case is that of three lobes as shown in Fig. 6. All of the radii center at the apices of an equilateral triangle, so that the sum of any two radii from an apex is a constant. If parallel measuring anvils are used, the diameter is the same in any position. For this reason it was some time before it was realized that the lobing existed. If, however, we measure from a V-block the error is at once apparent.

The lobing occurs during the rough-lapping operation. One-inch-diameter rolls usually have from 15 to 33 lobes, and the three-lobed form only appears when the diameter is less than $^1/_8$ in. The amount of deviation from the true cylinder is usually negligible, but can be felt as a roughness when the sagitta error is only 3 or 4 millionths of an inch.

Since the lobed pieces are identical in diameter, the removal of the lobing is easily done by a rapid hand operation with a ring lap. Several passes of the lap to and fro are sufficient to remove the bumps and, due to the identity of size, the lap is adjusted only once for the lot.

Once removed, the lobing does not appear spontaneously with further lapping. It returns, but creeps in so slowly that the diameter may be reduced as much as 0.0002 in. without introducing a measurable amount.

Measurement

To manufacture rolls in quantity to such precise limits as may be held with this method of lapping, we require refinements in measuring equipment which the standard measuring machine cannot give us. To this end the Pratt & Whitney Co. have developed two new machines which have proven entirely satisfactory in practice.

The millionth comparator shown in Fig. 7 is used as a transfer medium between the product and a master replica. It needs no comments here other than to state that it consists of a pair of parallel surfaces between which the comparison of a master size and a commercial replica is made. One of these anvil surfaces is movable axially, so that contact can be made with the measurement, and the axial displacement of the anvil can be read on an optical scale at a magnification of 20,000 to 1. This scale is comparatively coarse [0.5 mm. movement of the scale being equivalent to 0.000001 in. (1×10^{-6}) displacement of the anvil], so that measurements may be transferred from the master to the replica with less than 0.0000002 in. (2×10^{-7}) loss in accuracy or error.

We have found it impossible to use a master which is not a replica of the work where extremely accurate matching is required. For instance, if we set the machine on a 1-in. Hoke block and measure a 1-in. cylinder, the cylinder will appear small by two or three millionths, due to distortion of the contact. If we use a replica, however, the contact distortion is the same on the replica as on the work, and no accuracy is lost.

These master replicas or primary masters are calibrated on the interferometer. By means of this apparatus we can determine the number of light waves of known wave length (or color) which at a given instant are between two planes coincident with the opposite parallel faces of a gage block. When this number is known the length of the gage can be calculated. The light beam is therefore analogous to a scale with divisions approximately $^2/_{100,000}$ in. apart.

When a measurement is to be taken we choose two cylinders of identical size in the millionth comparator. These two rolls are then placed in a holder, as shown in Figs. 8 and 9, between two quartz planes which are held against the rods by light springs so placed that no bending moment exists in the planes. The planes are either dead flat or their errors have been studied and are known. Thus the position of two little patches of gold plate relative to the cylinders is determined.

When a light of known wave length is allowed to pass the planes and interfere, we determine the relation between the gold mirrors. This value corrected by the plane errors and contact distortion errors and for dispersion of the light, becomes the diameter of the rolls. The path of a split ray is shown as a heavy line in Fig. 9.

Fig. 10 shows a general view of the interferometer. The holder with the rolls and planes is inserted in a constant-temperature box. Temperature control is maintained by circulating oil through a series of coils and by means of a thermostat in conjunction with an electrical heating unit which serves automatically to maintain the temperature within very close limits.

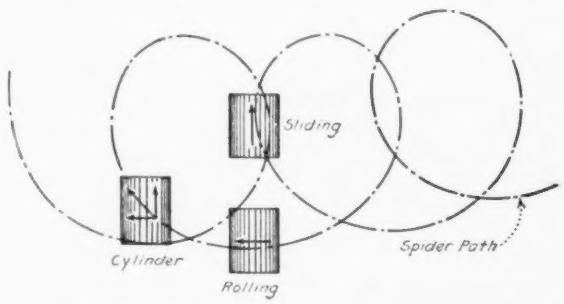


FIG. 4 PATH OF ROLLERS ON LAPS. HOKE MACHINE

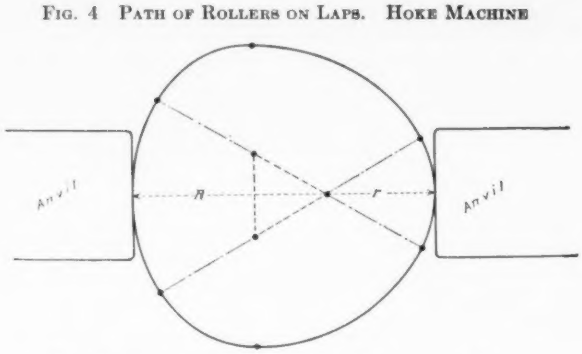


FIG. 6 THREE-LOBED ISOGON. DIAMETER = $R + r$

The standard wave lengths used are those emitted by glowing neon gas under electric discharge. These wave lengths have been calibrated against the Michelson-Benoit standard by Meggers and Peters of the Bureau of Standards to an accuracy of one part in four millions. It has been found to be a satisfactory and constant reference standard under easily reproducible physical conditions. Light waves from this source enter through a lens and suitable opening into the constant-temperature box, pass through the quartz planes referred to, and then through another lens and tube to a prism located within the cylindrical box. The light is refracted and the rays of different lengths separated by this prism so that they can be observed separately through the telescope.

The holder carrying the glass planes and rolls to be measured is attached to and turns with a long protractor arm of light tubular construction. At the outer end of this arm there is a scale indicating angular movements, which is enlarged by a small telescope. The angular position of the protractor arm and of the quartz planes is regulated by turning a handwheel which transmits motion to the arm through suitable shafting and gearing.

Now at certain angular positions of the quartz planes the phenome-

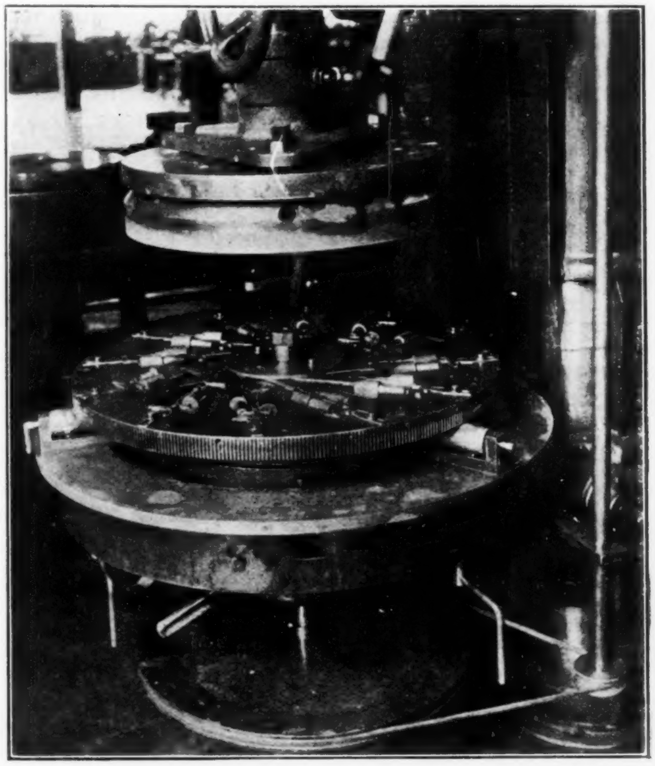


FIG. 5 ROTARY RIG OPEN

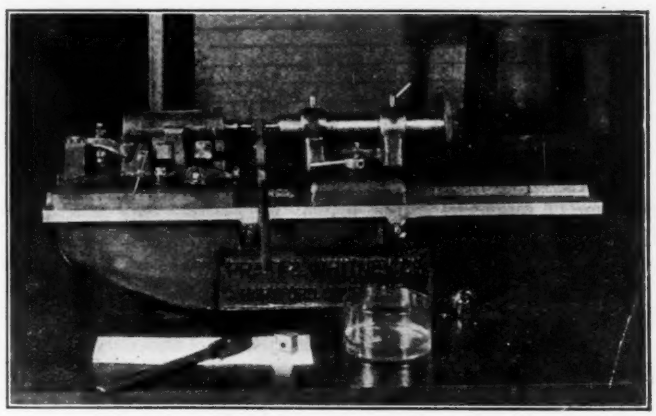


FIG. 7 THE MILLIONTH COMPARATOR
(Fundamental design by National Physical Laboratory, England.)

non of light interference occurs. These angles, which vary for different light rays, are noted by taking readings on the scale seen through the small telescope. When several of these angular values are known, in addition to the lengths of the different light waves, the required distance between the two planes is determined mathematically by a process of elimination carried on until the true value is obtained, as proved by the fact that it is the only value that satisfies an equation containing the other known values. When the number of waves is found (within a small fraction of a wave) it is multiplied by the known wave length, and the product is the absolute diameter of the roll. This gage is now a master, or primary standard, and may be confidently used to calibrate commercial gages in the millionth comparator.

Before a measurement can be taken, the roll must be left in the constant-temperature box until it has acquired a steady temperature. Also considerable time is needed for the various adjustments, readings, and calculations, so that the measurement of a roll would ordinarily require an entire day.

Reference to Fig. 11 will show where the extraordinary accuracy of the measurement arises. $N\lambda = 2E \cos \theta/2$ is the general equation for interference in a single color. Combining several equations specific to their separate wave lengths gives n simultaneous equations and $n + 1$ unknowns. From hypothesis we know that these unknowns must be integers. So we guess one integral value and solve for the remaining ones. When we have finally guessed correctly, the departure of $N_1, N_2, N_3 \ldots N_n$ from integrity is the experimental error, which can then from theory and hypothesis be wiped out.

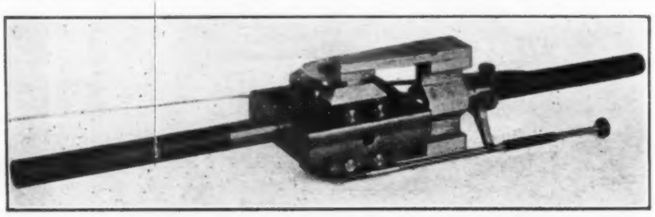


Fig. 8 Etalon for Cylindrical Measurement

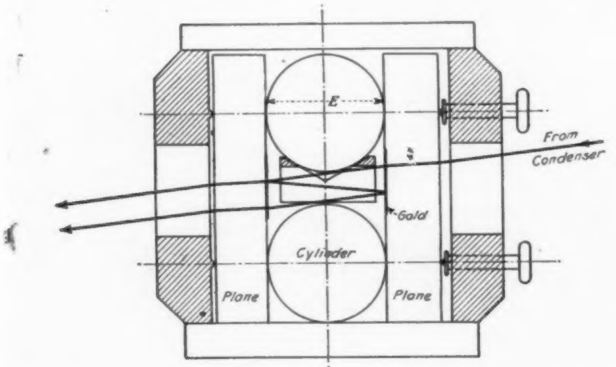


Fig. 9 Etalon for Cylinders

The remaining uncorrected errors which affect the measurement are:

a Calibration error of the light source

b Unavoidable and unknown distortion errors in the etalon

c Errors in measurement of the interfering angle.

The sum of this residuum is probably less than 0.4×10^{-6} in.

International Tests

A little over a year ago the Pratt & Whitney Co. decided on a test to determine if possible the most probable universal value of the inch. We made up 24 rolls as closely to size as we could determine with the interferometer and sent six to England to the National Physical Laboratory, six to the French Bureau, and six to the United States Bureau of Standards for calibration.

These rolls were to be intercompared and compared against the standards of the several countries and then returned, when they were to be shipped a second and a third time until each had a record from the several bureaus. The test has not been completed so far, but such information as we have shows remarkable agreement.

The author measured the rolls several times on the interferometer during the lapping until they were less than a millionth high. When first completed they were so nearly identical that the millionth comparator could not distinguish between them.

After about six months in storage they were measured again for secular change and were found to have changed less than 0.4 millionth. The average of the group was about 0.6×10^{-6} in. high.

The National Physical Laboratory reported on their first group a short time ago and the disagreement with the Pratt & Whitney reading was less than 0.8×10^{-6} in. The two reports were based on a light-interference method and a mechanical method, respectively. The dates of the Pratt & Whitney Co. and the National Physical Laboratory readings are six months apart, which shows that the conditions were severe.

If we trace through all the conversions necessary, the agreement seems all the more remarkable.

	Conversion	By
1	Metal to neon	P. & W.
2	Neon to cadmium	Meggers & Peters, B. of S.
3	Cadmium to meter	Michelson & Benoit
4	Meter to British line yard	French & English Bureaus
5	Line yard to end yard	N. P. L.
6	End yard to end inch	N. P. L.
7	End inch to P. & W. metal inch	N. P. L.

Error, 8 parts in 10 millions, equivalent to increasing the diameter of the earth about 35 ft.

To recapitulate, any round piece can be lapped in quantity to a highly finished and accurate surface with speed.

Where the conditions demand, extreme precision can be maintained with practical equipment and controls.

The apparatus for lapping is not particularly expensive, nor is it difficult to maintain and operate.

From the gage-user's viewpoint, the chief value of the method is its inherent ability to cheapen overall gage cost.

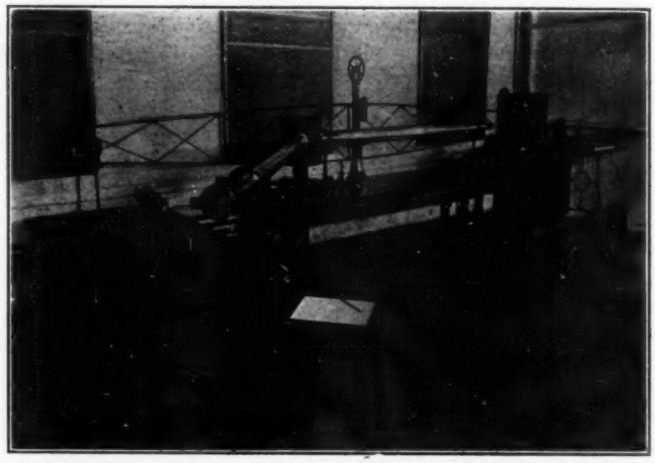
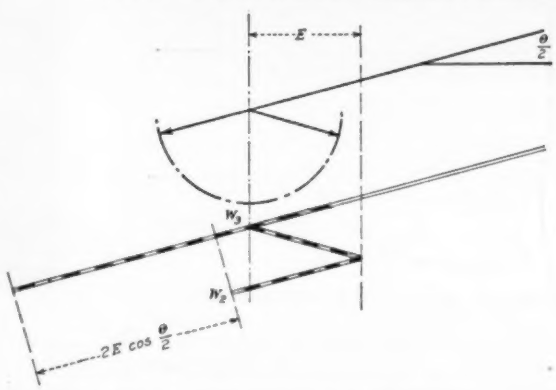


Fig. 10 Interferometer—Modified Fabry and Perot Arrangement



$$P\lambda = 2E \cos \frac{\theta}{2}$$

P = number of waves

λ = wave length

Let $P = N + a$

N = integer

a = fraction

$$2E = \frac{(N + a)\lambda}{\cos \frac{\theta}{2}}$$

If we observe on bright fringes,

$$2E = \frac{N\lambda}{\cos \frac{\theta}{2}}$$

Fig. 11 Interference Calculations

$$2E = \frac{N_s\lambda_s}{\cos \frac{\theta_s}{2}} = \frac{N_1\lambda_1}{\cos \frac{\theta_1}{2}} = \frac{N_2\lambda_2}{\cos \frac{\theta_2}{2}} = \frac{N_3\lambda_3}{\cos \frac{\theta_3}{2}}$$

$$N_s \left(\frac{\lambda_s}{\lambda_1}\right)\left(\frac{\cos \frac{\theta_1}{2}}{\cos \frac{\theta_s}{2}}\right) = N_1 \qquad N_s \left(\frac{\lambda_s}{\lambda_2}\right)\left(\frac{\cos \frac{\theta_2}{2}}{\cos \frac{\theta_s}{2}}\right) = N_2 \qquad N_s \left(\frac{\lambda_s}{\lambda_3}\right)\left(\frac{\cos \frac{\theta_3}{2}}{\cos \frac{\theta_s}{2}}\right) = N$$

$N_s = 38{,}620.000$ $\qquad$ $N_1 = 45{,}729.012$ $\qquad$ Thus $N_1 = 45{,}729.000$

High-Speed Cutting of Brass and Other Soft Metals in Standard Machine Tools

Types of Machines to Use—Tools and Tooling Methods to Employ—Cutting Dry or with Coolant—Kind of Metals to Use—Examples of Work Done in the Screw Machine

By LUTHER D. BURLINGAME,[1] PROVIDENCE, R. I.

CHALLENGING questions relative to operating on soft metals bring such varied answers and the expression of such diverse opinions that the conclusion must be reached that there is as yet no standardized practice. This seems to be especially so in making use of general-purpose machine tools and adapting them to high-speed work of this character.

Some of the questions raised are: (1) The type of machine suited for the work; (2) the kinds of tools and methods of tooling to be used, including whether carbon or high-speed steel should be used; (3) the kind of coolant, or the question of cutting dry; (4) the kind of metal to be used, the comparative cost between steel and brass, variation in cutting qualities of soft brass; and (5) the skill of layout men and operators.

Type of Machines

Many factors must be taken into consideration in determining the machine to use and the most efficient means to obtain the maximum production.

Essentials are that it shall be possible to run the spindle at high speed and to secure proportionally increased feeds. Tools must be so rigidly supported that they will stand the higher feeds and speeds and still produce smooth and accurate work without the necessity of too frequent grinding.

The higher the rate of production, the more important becomes the saving of idle time. This is noticeable not only in screw-machine but also in milling-machine work. In semi-automatic machines, where loading can be done simultaneously with the cutting and the time of loading can be kept within the time of cutting, this portion of the idle time can be eliminated.

In a machine made primarily for high-speed work such as the B. & S. No. 19 automatic screw machine, Figs. 1 and 2, the regular spindle speed is 5000 r.p.m. Distinctive features of this machine were described in an article by the author in *Machinery* for March, 1923, under the title, Screw Machine Products Made in Less than Three Seconds. Fig. 2 is a section through the spindle showing the method of lubricating, circulating a coolant around the bearings, supporting the driving pulleys independently of the spindle, etc.

Drilling or other auxiliary speeds can be increased materially above this by running the drill spindle in the opposite direction at 3350 r.p.m. giving a net speed of 8350 r.p.m. For tapping and threading, the differential speed obtained by running the tap or die in the same direction as the spindle is made use of.

The timing of the cams on this machine is such that the cycle of operations can be completed in two seconds, but that does not mean that this is the minimum time for completing a piece, as it is possible to use more than one set of lobes on the cams and in such cases to complete two or more pieces at each revolution of the driving shaft. The indexing of the turret can often be avoided, thus reducing idle time. The use of a swing stop when feeding stock makes unnecessary the use of a turret station for this purpose.

Much might be said as to the comparative merits of single- and multiple-spindle machines for this type of work—each having advantages under certain conditions—taking into consideration the relative investment, floor space, idle time for changing the work, repairs, etc.

On the regular line of B. & S. screw machines provision is made for operating on soft metals by the substitution of a spindle designed for high-speed work which runs at materially greater speeds than the spindle regularly provided. This is done by using roller bearings, ball thrusts, and special provisions for oiling. Further time can often be saved by speeding up the driving shaft by a redesign in clutches and changes in feeding mechanism so as to reduce the time of idle movements. Fig. 3 shows a No. 00 automatic screw machine equipped for high-speed work.

As the full-automatic machines of this type require a reverse for threading, they cannot be operated when using the reverse at as high a speed as that used for machines of the non-reversing type because of the shock of reversal. They can, however, be speeded up to this maximum speed, so as to be available when reversal is not required. Taking the 00 size as typical, which runs normally at a maximum spindle speed of 2400 r.p.m., this can with the high-speed spindle be satisfactorily speeded up to 5000 r.p.m. when used without reversing, thus more than doubling its speed, and up to 3600 r.p.m. when the reverse is used.

The corresponding turret forming and cutting-off machines, where no reverse is provided, are run, for work on soft metals, at 5000 r.p.m. The Nos. 0 and 2 sizes of machines can be speeded up proportionately.

The use of attachments makes it possible to perform many operations which would otherwise make necessary a second handling of the work in another machine. As such operations are usually coincident with the primary operations, no extra time is required.

Attachment work includes milling, cross-drilling and gear-cutting

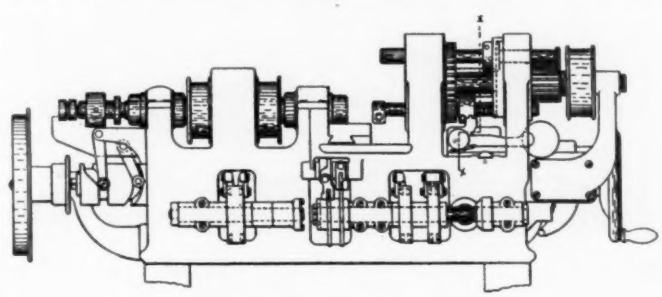


Fig. 1 No. 19 Automatic Screw Machine
(Adapted for high-speed work. Tool carriers of light construction and independently traversed.)

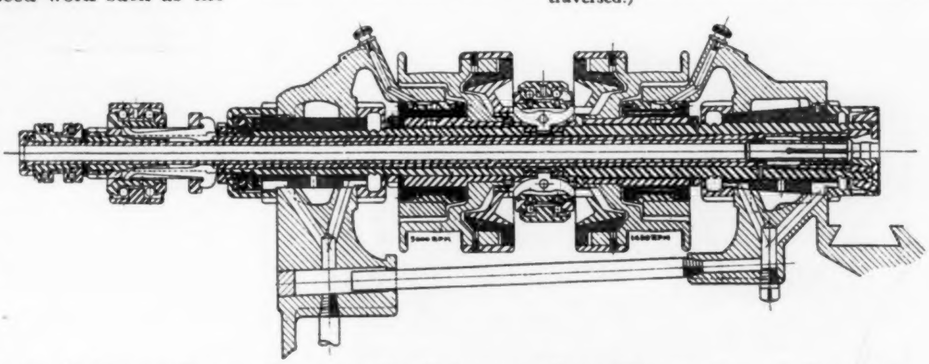


Fig. 2 Section Through Spindle of No. 19 Automatic Screw Machine, Showing Construction Suited to Operating at 5000 R.P.M.
(Pulley bearings independent of spindle; circulation of oil for spindle bearings.)

[1] Industrial Superintendent and Patent Expert, Brown & Sharpe Mfg. Co. Mem. A.S.M.E.

Contributed by the A.S.M.E. Machine Shop Practice Division for presentation at the New Haven Machine Tool Exhibition, Mason Laboratory, Yale University, New Haven, Conn., September 8 to 11, 1925.

operations, Fig. 4 being an illustration of a milling operation while Fig. 5 shows an attachment for cutting small worms.

Hand milling machines, such as those built by the Pratt & Whitney Co. or the small-sized automatic or semi-automatic milling machines now on the market but specially speeded up for this work, are well adapted to such needs. The No. 0 plain milling machine using a high-speed horizontal spindle, as shown in Fig. 6, or a high speed vertical spindle as shown in Fig. 7, can be equipped with a semi-automatic control, Fig. 8, so that the operation of a foot treadle brings the work to the cutter, at which time automatic engagement of the feed takes place and at the completion of the

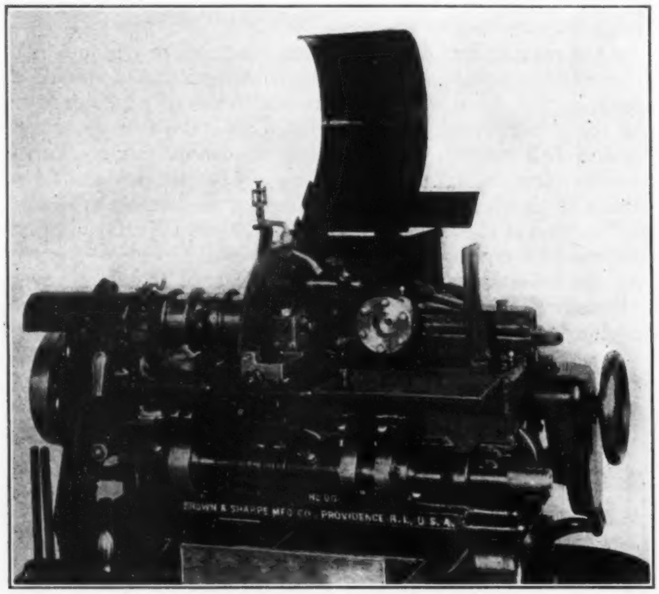


FIG. 3 SPECIAL No. 00 AUTOMATIC SCREW MACHINE SUITED TO HIGH-SPEED WORK
(With spindle which can be run at 5000 r.p.m., driving shaft speeded up, double indexing, swing stop. Special oiling for clutch body.)

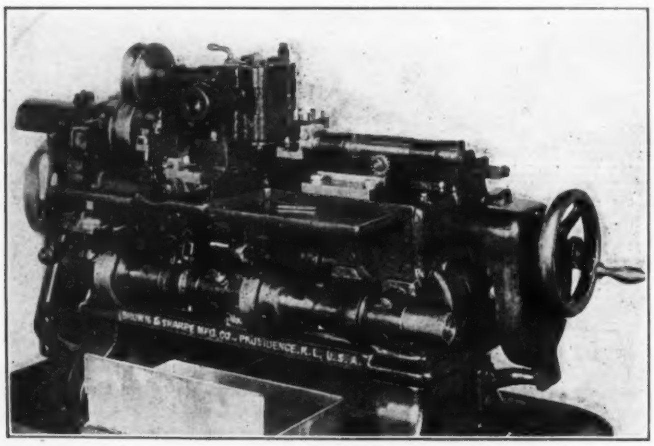


FIG. 4 DOUBLE MILLING OPERATION ON B. & S. AUTOMATIC SCREW MACHINE
(Skiving tools used to finish aluminum piece.)

cut a trip operates so that the table automatically returns to the loading position. The use of the foot treadle leaves the hands of the operator free to load and unload the fixture.

Tools to Be Used

Generally speaking, the same styles of cutting tools, but provided with less clearance, are used when cutting soft metals as when operating on steel. This is not always the case, however, as rolling threads on soft metals is a common practice and sometimes can be used to marked advantage. It is possible also to use skiving methods, especially effective when operating on aluminum.

As to the comparative use of carbon- and high-speed-steel tools, there seems to be a difference of practice, there being those who claim that carbon-steel tools can be used with equal success as compared with high-speed-steel tools when cutting on soft metals, thus saving the greater expense of the latter. The highest cutting speed in the examples here illustrated is in a milling operation (Example 21) operating at 1050 ft. per min.; in this case carbon-steel cutters are used, these not requiring to be sharpened oftener than once in two days, even when cutting dry. In the case of many screw-machine jobs the diameter of the work is so small that the surface speed of the cut is below the maximum, so that the cheaper carbon-steel tools can be used without question. Where high-speed-steel tools are used without grinding they may not give as smooth a finish as will carbon-steel tools.

The Kind of Coolant—Cutting Dry

While much work in cutting soft metals, especially in milling, is done without the use of a coolant and tools will stand up surprisingly well under such conditions, the general practice is to use some coolant, usually lard oil. Even those who make a practice of using lard oil exclusively, however, admit that it is not only much more expensive but is not as satisfactory as a coolant as some of the soda-water preparations. The difficulty in the use of soda water is its penetrating to the bearings and getting between the slides of the machine, resulting in hard action and breakage, a difficulty which does not exist when oil is used. There are, however, those using special soda-water solutions as coolants who claim that their mixture is such as to avoid this difficulty and give them the benefit of low cost combined with a more satisfactory coolant.

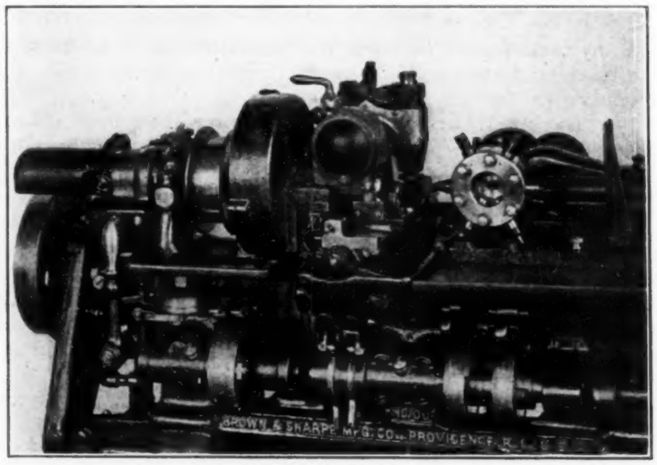


FIG. 5 ATTACHMENT FOR CUTTING THE THREADS OF WORMS BY THE HOBBING PROCESS ON B. & S. AUTOMATIC SCREW MACHINES

A coolant which gives good results in machining aluminum is made by mixing equal parts of kerosene and lard oil.

The Kind of Metal to Be Used

Although steel costs but approximately one-third as much as brass, the much greater speed possible in machining the latter often makes possible such an increase in production that the question of initial cost of the stock is not only offset but a material reduction in cost results from using the more expensive metal. Examples of such a saving were shown in an article by H. L. Burlingame in the *American Machinist* of June 7, 1923, and in the issue of Feb. 5, 1925, Sydney Fisher of the Bridgeport Brass Co. gives rules and shows diagrams to aid in determining the point at which it becomes more economical to use brass instead of steel.

An important factor to consider is the salvage of scrap, which is so much greater for brass than for steel that often the labor cost of making brass parts can be paid for out of the sale of the scrap.

Most of the pieces shown in Fig. 9 are made not only at a much lower direct cost than if made of steel, but there is also a material saving in the floor space and overhead charges because of the fewer machines and the shorter time required. For example, No. 3, if figured on the same basis of machine time and man time as in the *American Machinist* articles referred to, could be produced more than six times as fast when made of brass as compared with

steel, and the pieces when made of steel would cost more than 2.35 times as much as if made of brass.

Due to increased output and the extra work of supplying a larger number of rods of stock and in caring for the chips, the labor cost of operating on brass might be slightly higher than for steel. This should be balanced, however, by the longer life of the tools, when cutting brass, as they require less frequent grinding and setting.

One of the questions in determining the cost of making parts of brass is the quality of the brass stock used. It is claimed that there is a material difference in the working qualities of various brands of soft brass. For example, brass rods made abroad are as a rule harder and do not work as readily as similar stock made in this country. This means that in setting up jobs to work on the harder brass, slower feeds and speeds may be necessary and the use of high-speed-steel tools called for.

It is economical to use tubing for some work made on the screw machine, although tubing as a rule is somewhat harder than brass rods; and this must be taken into consideration in determining on

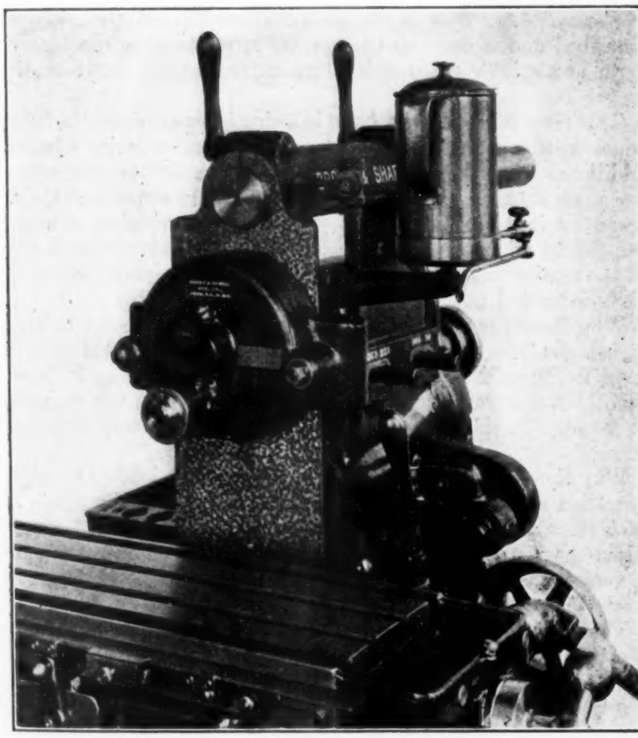


Fig. 6 No. 0 Plain Milling Machine with Spindle Speeded Up 4-1, Giving 1840 R.P.M. When Running at Regular Speed

(Suited to the use of end mills and for general use in cutting soft metals.)

the economy of its use, as well as the higher cost per pound of the stock.

Skill of Layout Men and Operators

Where there are so many conditions involved and where the variety of work is almost infinite, long experience and special skill in devising the best ways of doing the work and determining on the machines where it can be most economically handled are of greatest importance. It is sometimes found that even after expert tooling up of a job which seems to be the last word in high-speed production, another method can be suggested which will give higher production and as good if not a better quality of work. The alertness and skill of the operator in keeping his machines going and in showing an interest to bring out the maximum production are likewise determining factors in securing satisfactory results. It is customary to allow a reduction of 10 per cent from the maximum continuous output to provide for the ordinary contingencies which arise, although it will be understood that this amount varies, depending on the conditions.

Examples of Work Done on the Screw Machine

The following examples of work on brass are actual parts which have been practically run, and while it will be seen that production in some cases could be increased if different equipment were used, they will nevertheless illustrate actual conditions the more fully, because a user does not always have the equipment which will give

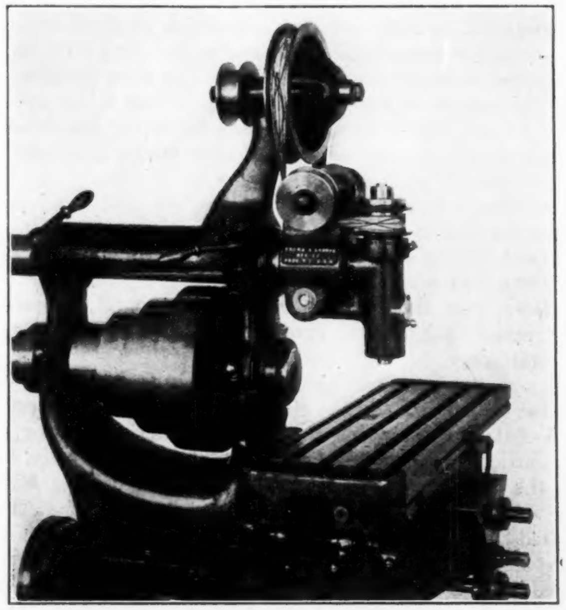


Fig. 7 High-Speed Vertical Spindle Attachment for Use on No. 0 Plain Milling Machine

(Can run at 2000 r.p.m. of spindle, or by use of ball bearings, at 4000.)

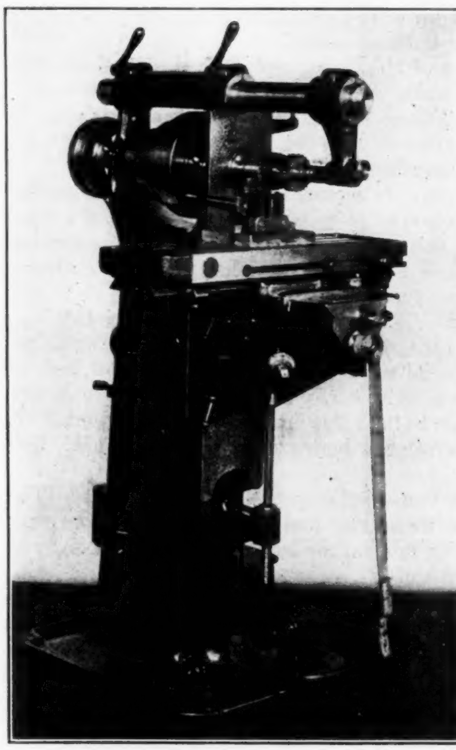


Fig. 8 No. 0 Plain Milling Machine Suited to High-Speed Production

(Equipped with semi-automatic control so as to leave the operator's hands free to place and remove the work.)

the maximum efficiency for every one of a number of different jobs; and, as already pointed out, the limited quantity of parts may not warrant the installation of the most efficient equipment for some unusual job, and a general-purpose machine will have to be used which has been built to cover the range of a great variety of work and be adapted as best it can to the particular job in hand.

The examples illustrating the use of attachments on soft metals show that in most cases there would be no material advantage

in speeding up the attachments as they operate simultaneously with the machine operations and the latter require a greater length of time. For high-speed work an aluminum arm can be substituted to advantage for the ordinary cast-iron transfer arm.

Example 1 includes the operations of knurling and drilling. As no reversal of spindle or turret indexing is required, this job can be done on a cutting-off machine, and as the diameter is but $^9/_{64}$ in. it is suited to the 00 size of machine. This is an instance of using a special spindle at a speed of 7000 r.p.m. and a high-speed drive shaft; a swing stop is used for feeding the stock; the surface speed of stock is 262 ft. per min. The time required to complete the piece is but 0.9 sec.

Could not a turret drilling attachment be used to advantage in drilling the hole in this piece? The piece is so tooled as to require a support on the stock diameter while knurling. The drilling is also being done while knurling, and while it is held in the support. Therefore, were the drill to be revolved it would be necessary to have special equipment to revolve the drill while the support was held stationary.

Example 2. The small pin shown in this example could be made of either steel or brass in the given time because the highest surface speed obtainable is within the range of practical cutting of steel. This particular job was set up for the 00 automatic screw machine with the spindle running at 5000 r.p.m. and using a high-speed drive shaft, giving a surface speed of but 82 ft. per min., the time required being $^3/_4$ sec. If this job were done on the same machine and at the same speed as Example 1, it could be done in $^1/_2$ sec.

Could not a coarser feed be used on cut-off? No. On small diameters the feed must be very fine or the piece will break off rather than be cut off.

According to the change-gear table furnished with this machine, the fastest time possible to make one piece when the driving shaft is speeded up is $1^1/_2$ sec. However, the piece is produced in $^3/_4$ sec. How is this possible?

The time of $1^1/_2$ sec. represents the fastest time in which the camshaft can make one complete revolution. However, if two pieces are made in one revolution of the camshaft, this means that a piece is made every $^3/_4$ sec. In order to do this it is necessary to have duplicate sets of lobes on the cams.

Example 3. This piece is made on the 00 automatic screw machine with a spindle speed of 5000 r.p.m. and a high-speed drive shaft. In this case the thread is cut with an opening die so that the reverse is not required. The maximum cutting speed is 572 ft. per min.; time required, 5 sec.

How can $^7/_{16}$-in.-diameter stock be run in the No. 00 machine when the capacity of the oversize feed tube is only $^3/_8$ in.? By using a special soldered-in feeding finger and feed tube.

There is a limit of only 0.0005 in. in the hole on this job. How is it possible to hold this limit at such high speeds? Following the drilling operation a boring operation is provided, then a reaming operation.

Why are two spindle speeds used on this job? The slow speed is used for threading; and this allows for more cam surface on the threading lobe on the lead cam; this is especially advantageous on short threads.

How are these two speeds obtained? By driving both spindle pulleys in the same direction but at different speeds and by throwing the clutch body from one pulley to the other. This is the same mechanism that reverses the spindle on jobs where button dies are used. By using a slower speed for threading, the die heads operate better, cut a smoother thread, and stay sharp longer, consequently producing a more accurate thread.

Example 4. An example having a threading operation which in order to be produced without a second handling must have the thread rolled. This is a job done on the No. 00 automatic turret forming machine with the spindle speed at 5000 r.p.m., but without the fast drive shaft. It is not practical to use the fast drive shaft on this job because the turret traverse must then be reduced to $^5/_8$ in. so that it is not practical to drill deeper than $^1/_2$ in., and the hole in this piece is $^5/_8$ in. deep.

This gives a surface speed for forming of 500 ft. per min.; time, 5 sec. The thread is rolled by a top-thread roll holder, held in conjunction with the cut-off tool.

What are the advantages of rolling threads? Thread rolling avoids the reversing of the spindle and makes it possible to complete many parts which otherwise would have to have a second handling. For instance, were this piece made from steel it would be necessary to rechuck it to perform the threading operation.

Example 5-a shows the first operation on a piece requiring two handlings, this being done in a No. 0 automatic turret forming machine with a high-speed spindle at 3600 r.p.m., but with the regular drive shaft because the reduced traverse of the machine with speeded-up drive shaft would not be sufficient for this operation. The surface speed for turning is 530 ft. per min.

Why are not two speeds forward used on this piece, one for threading, as on some of the previous jobs? The threaded portion is quite long, which allows enough cam surface to develop the thread lobe on the lead cam without changing the speed; some time would be lost also on account of the length of the thread if the slower feed were used for threading.

Would it be practical to roll this thread? Not in a screw machine; the thread is too long. The pressure would be so great that it would spring the work and a full thread would not be rolled.

Example 5-b. This is the second operation on the piece just described and is done on the No. 00 turret forming machine. It is run at a spindle speed of 5000 r.p.m., feed of drill, 0.015 in., time, 5 sec.

This piece is inserted by hand in a circular magazine attachment, which in turn transfers it to the spindle of the machine where the additional operations are performed. Because the diameter of the stock is $^9/_{16}$ in. on this piece, it is necessary to perform the first operation in a No. 0 machine. However, as the piece is held on the thread diameter for the second operation, advantage is taken of the smaller machine with the higher spindle speed, which thus gives increased output.

Why should not a turret drilling attachment be used to obtain higher speed for this drill? The diameter of drill would be rather large for the size of this attachment. As the drilling is done in slightly over one second because of the coarse feed, little time could be saved. The idle time on this operation is 68 per cent of the total time.

Why should the idle time be so great on this job? On second-operation work it requires idle time to bring the piece up in line with the machine spindle from the magazine attachment, time to insert the piece and close the chuck, also time to open the chuck and eject the piece. The more turret operations, the larger the percentage of idle time.

The factors to be taken into consideration that go to make up idle time are the feeding of the stock, the indexing of the turret, etc. Idle time plays almost as important a part in determining the total time consumed as the cutting time, and therefore it is just as important to speed up the idle as well as the cutting movements. This is done on the screw machine by increasing the driving-shaft speed as well as by the use of the swing stop and double indexing. These features were explained in detail in an article in *Machinery*, September, 1923, entitled High-Speed Spindles for Screw Machines.

To illustrate the advantage of a fast drive shaft on screw machines adapted to use on soft metals, the actual time for indexing the turret of the 00 size machine is $^1/_2$ sec. By means of the fast drive shaft this time is cut to $^1/_4$ sec.

Example 6. These nuts are made on the No. 0 automatic screw machine with the high-speed reversing spindle at 2700 r.p.m. and using the fast drive shaft with double indexing and the swing stop; time, $3^1/_6$ sec.

These nuts are made by the double method; in other words, one cut-off tool is used on each cross-slide and two pieces are completed at each revolution of the cam shaft.

Is it practical to use this double method on many jobs? On nuts and washers or other thin parts that have long cut-off operations, considerable time is saved.

Example 7 requires the use of the No. 2 automatic screw machine. The spindle for this size of machine is speeded up to 2400 r.p.m. and use is made of the fast drive shaft; surface speed, 708 ft. per min. for the forming; feed for drilling, 0.012 in., for cutting off, 0.001 in. The threads in this case are rolled, and the time to complete the piece is 13 sec.

This job is ideal for rolling the thread, as in making it with the

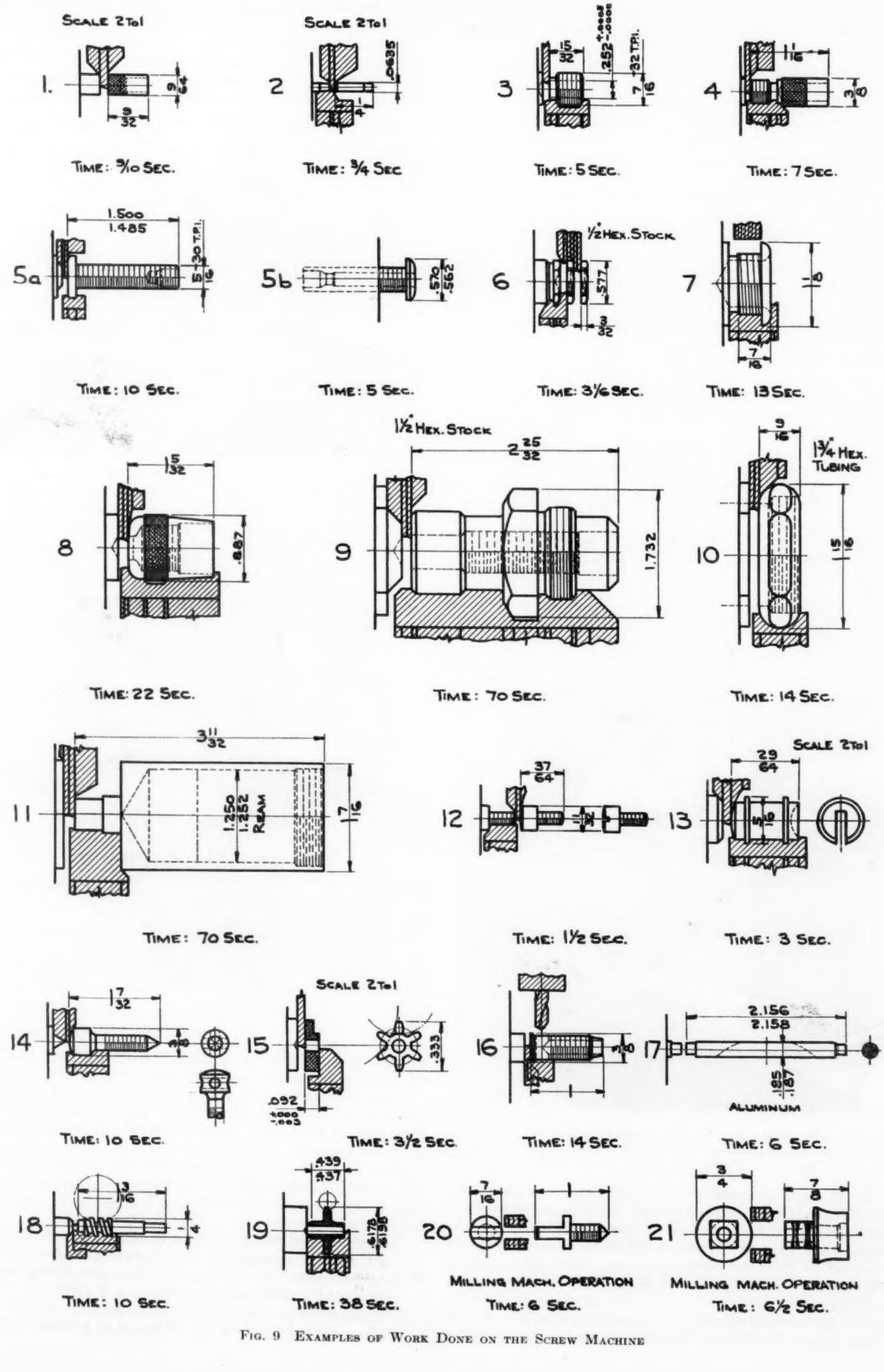


FIG. 9 EXAMPLES OF WORK DONE ON THE SCREW MACHINE

thread at the back it allows the inside radius to be put on the front of the piece by a turret operation and avoids the necessity of reversing the spindle when cutting the thread.

Why is the feed on the cut-off tool so slow on this piece? The cutting off is combined with the forming, and consequently the width of cut slows down this operation.

Example 8 is another job requiring the No. 2 automatic screw machine with high-speed reversing spindle, in this case at 1800 r.p.m. and with the regular drive shaft; time to complete piece, 22 sec.

Numerous operations are required to finish this piece. Knurling is done by a turret knurl holder.

Why is not a high-speed drive shaft used for this job? It could be used, in which case an additional saving of 6 sec. would be made, bringing the time down to 16 sec.

Example 9. This piece being made of $1^1/_2$-in. hexagonal stock, requires the No. 6 automatic screw machine for its manufacture. The machine is run at three speeds, 875 and 292 r.p.m. forward, and 167 backward. Time required, 70 sec.

In running hexagonal stock of this size trouble may be experienced if higher speeds are attempted in keeping the corners from being marred, besides making a great deal of noise in the wire stands, especially if the bars of stock are of any great length and are not quite straight.

Could not a turret drilling attachment be used to advantage, so as to obtain a higher cutting speed for the drill? Little would be gained, as the drilling operation overlaps the forming and the actual time consumed is that required for the forming operation and not the drilling.

Were there no forming on this piece, considerable time might be gained by the use of such an attachment.

Example 10. This is made of special drawn hexagonal tubing

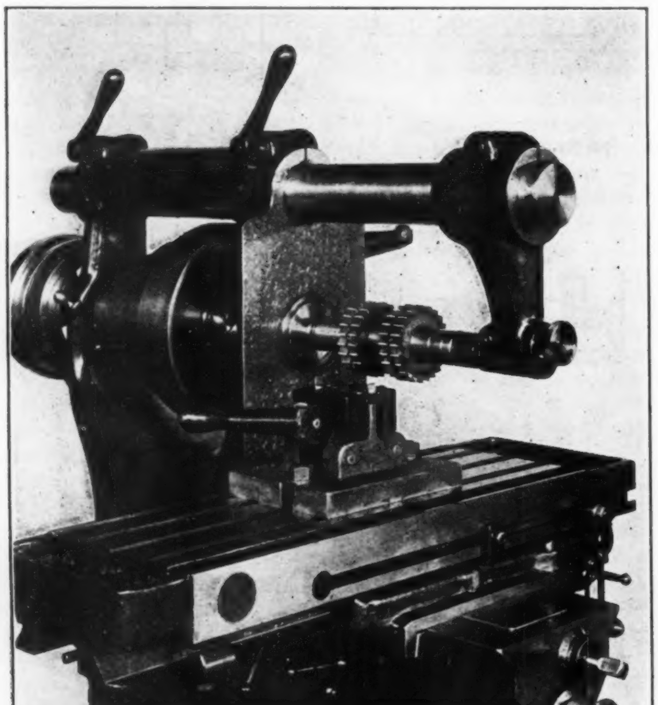


FIG. 10 NO. 0 PLAIN MILLING MACHINE WITH SEMI-AUTOMATIC ATTACHMENT FOR USE IN QUANTITY PRODUCTION AND WITH FIXTURE FOR MILLING "ARGAFFES" FOUR AT A TIME

and is run at a spindle speed of 875 r.p.m. without reverse; time, 14 sec. The use of a collapsing tap avoids the necessity of reversing the spindle.

Example 11 is done on the No. 4 automatic screw machine at spindle speeds of 1000 and 370 r.p.m. forward and 225 backward; time, 70 sec. The reamed portion of the hole must be very smooth and accurate as to size.

Would it be practical to use a turret drilling attachment on a job of this type? No; the hole is too large and it would require a great deal of power to drive a drill of this size. Turret drilling attachments are advantageous in drilling holes small in comparison to the diameter of the stock.

Example 12. This and the following group of examples illustrate attachment work where secondary operations are performed without rehandling and simultaneously with the major operations. This

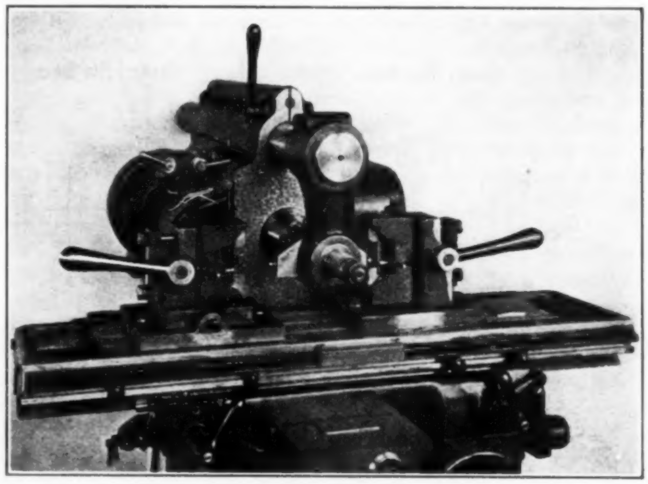


FIG. 11 NO. 21 AUTOMATIC MILLING MACHINE
(Work can be machined in one fixture while the other is being unloaded and reloaded, thus giving practically continuous operation.)

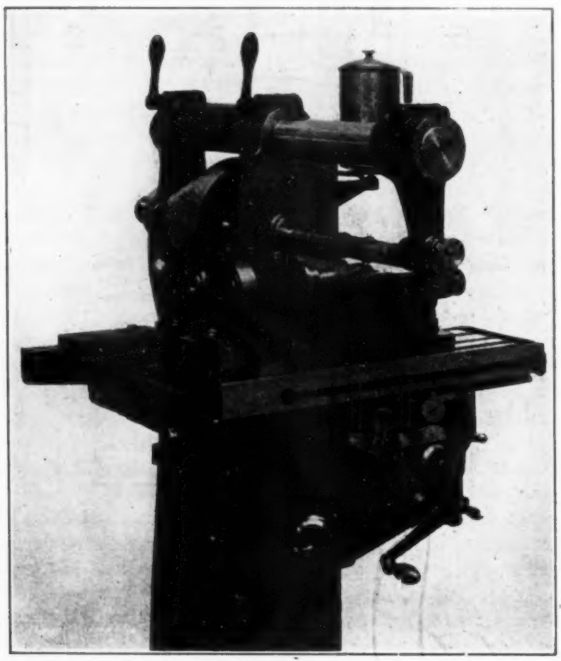


FIG. 12 CUTTING 108 BRASS GEARS AT A SETTING ON A NO. 0 PLAIN MILLING MACHINE IN $5^1/_2$ SEC. EACH

example is of a slotted screw made in the No. 00 automatic screw machine with a spindle speed of 5000 and 3600 r.p.m. forward.

The time required is $1^1/_2$ sec. As an opening die is used it is not required to reverse the spindle. This piece is made by the form and cut-off method, a screw being formed while the previous one is being cut off.

Does the screw slotting increase the time required to complete this piece? No; however, on some jobs it is necessary to allow time to get the pick-up arm in and to transfer the piece to the attachment.

Example 13 illustrates a milling operation which must be handled in quite a different way from the operation of slotting a head. The regular No. 00 automatic screw machine is used with the regular spindle speed of 2400 r.p.m. forward; time, 3 sec.

This job requires a special milling attachment, it being transferred from the spindle by a pair of fingers to a chute, from which

it is carried on to an endless chain with cogs which feed it past the cutter.

Would it not be possible to do this piece to advantage on a machine equipped with all the high-speed features? Yes. The production could be about doubled. This would mean that the speed of the cutter in the attachment would have to be increased to give the best results. This cutter is now running at approximately 300 ft. surface speed.

Example 14 shows index drilling as done on the No. 00 automatic screw machine with the regular speeds, 2400 forward and backward. Time required, 10 sec.

The time on this job could be reduced if a high-speed spindle and an opening die were used. It would be impractical to use the fast drive shaft, however, as the length of the piece is beyond the capacity of the turret travel on the No. 00 machine so equipped.

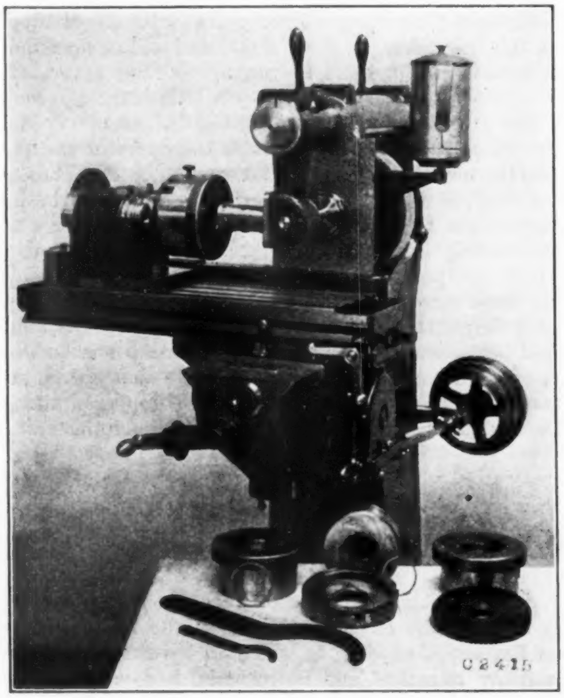


Fig. 13 Cutting 36 Internal Gears at a Setting on a No. 0 Plain Milling Machine in 20 Sec. Each

Index drilling is preferable to cross-drilling as it can be done without taking extra time and without stopping the spindle.

Example 15. This gear is made from extruded stock, the special feature of interest being that every second tooth is cut away for part of its depth, which makes a triple end-milling job as a feature of screw-machine work. This is done on the No. 19 automatic screw machine; time required, $3^1/_2$ sec.

This attachment is unique in that it consists of a head holding three small cutters and is clamped in the turret of the machine. Two pieces engage slots in the spindle-chuck nut, and this drives the cutters through gearing at a very high rate of speed, giving 325 ft. per min. surface speed. The lead cam feeds the cutters into the work to the proper depth. On this machine the drill speed is 8350 r.p.m., the spindle speed plus the drilling-attachment speed.

Example 16 illustrates both cross and longitudinal drilling on the No. 00 automatic screw machine. The surface speed of forming and turning is 354 ft. per min.; time, 14 sec. The speed of the attachment has been increased in proportion to the increase in spindle speed.

Example 17 has the unique feature of double milling on aluminum a slot in the side and another across the end of the piece. This is done in the No. 00 automatic screw machine by the use of a special two-spindle milling attachment mounted on the front cross-slide. Fig. 4. The two slots are milled simultaneously. Time, 6 sec.

It must be remembered that the screw machine was not designed as a milling machine, but when milling operations can be performed at the same time as other operations and at the one handling, often a great saving is effected.

While in the main aluminum can be operated on at speeds and with tools the same as for soft brass, its use involves certain problems such as caring for the chips, which are more troublesome because they do not so readily free themselves and thus load the machine up. This metal seems stringy and has a tendency to tear, so that it is difficult to obtain a smooth cut.

Example 18 is for producing a piece having a worm thread upon it. This attachment, Fig. 5, is mounted on the rear cross-slide and the cutter or hob is fed into the work by the cross-slide cam. Time required, 10 sec.

Example 19 is a wormwheel made on the No. 0 automatic screw machine with a spindle speed of 1800 r.p.m. The teeth are cut with a special hobbing attachment, mounted on the back cross-slide. Time, 38 sec.

The attachment used in cutting these small wormwheels is very similar to the attachment used for cutting the worms illustrated in Example 18. It will be noticed that whenever the cross-slide is used for an attachment the cut-off tool is usually combined with the forming, and this necessitates a slower feed on this operation.

Operations in the Milling Machine

Example 20. This piece is made on the screw machine up to the point of milling the flats. An illustration has already been shown, Fig. 8, of a No. 0 plain milling machine using a semi-automatic attachment; a close-up view of this same machine with the fixture for milling is shown in Fig. 10. With the spindle running at 700 r.p.m. 500 pieces per hour can be produced on this machine. The direction of the cutters is such as to feed on to the cut, and the cutting is done dry. In this case the cutters are made without side clearance, in fact, with a slight drag to the teeth so as to produce a smooth finish, practically without a burr.

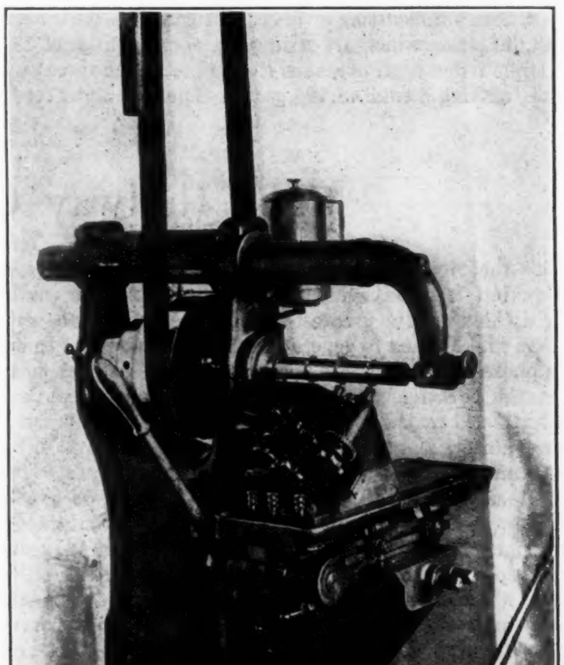


Fig. 14 Cutting Three Bevel Gears at a Setting on a No. 00 Hand Milling Machine in 22 Sec. Each

This same piece, milled on the No. 21 automatic milling machine running at the same spindle speed as above can produce 600 pieces per hour, this being an automatic machine in that after loading it has a quick traverse to the point of cutting, a slowing down for the cutting feed, reversing and returning at a quick traverse to the point where it engages a piece which has been placed in the opposite fixture and is in turn ready to be operated upon, the spindle having been automatically stopped during the return and started in the

opposite direction for the succeeding cut. In this case the loading is done during the time of cutting, and the difference of 100 per hour in the number of pieces between the product of this machine and the machine previously referred to represents the time thus saved. Although cutting dry, 45,000 pieces can be machined between grinds when using high-speed-steel cutters.

Fig. 11 gives the general appearance of the No. 21 automatic milling machine as set up for such a job. Offset fixtures are often used with double sets of cutters, so that by reversing the spindle it operates to cut in the same direction on to the work whether at the right or left.

Example 21 is a piece requiring indexing so as to mill the four sides. This job is done on a Pratt & Whitney hand milling machine, using two carbon-steel milling cutters $2^3/_4$ in. in diameter, at 1500 r.p.m., and 500 pieces (1000 cuts) are produced per hour. In this case the loading time is greater than the milling time, as the pieces are fed through by hand practically as fast as the operator can push the cutter through.

Gear-Cutting Operations on Milling Machines

By providing special attachments, manufacturing jobs of cutting gears of many kinds can be successfully done on the plain milling machine. Small plain milling machines which can have the spindle speeded up or be provided with high-speed attachments such as have already been described and shown in Figs. 6 and 7 are found useful for such work.

Fig. 12 shows the cutting of brass spur gears $^1/_{16}$ in. thick having 42 teeth, 30 pitch, which are strung together in gangs of 36. The use of triple index centers makes it possible to cut three strings at one time, making a total of 108 gears. The automatic feed of the machine at 0.030 in. per revolution is used for this job, but the return motion is by hand. After the cutters are clear of the work the index operates automatically by means of a cam plate back of the table. The cutters are $2^1/_4$ in. in diameter and run at a spindle speed of 1000 r.p.m. giving 580 ft. per min. cutting speed. The time for completing each gear is $5^1/_2$ sec.

Fig. 13 shows the cutting of internal gears on the No. 0 plain milling machine by the use of a special internal gear-cutting fixture, the gears in this illustration being $^1/_{16}$ in. thick and having 62 teeth, and 36 gears are cut at each loading. The cutter spindle speed is 1000 r.p.m., giving a surface speed of 580 ft. per min.; the feed per minute is 0.030 in., and the time required for each gear is 20 sec. Indexing is performed as in the previous example.

Fig. 14 shows the cutting of bevel gears by the use of triple index centers, this being done on the No. 00 hand milling machine. The cutters in this machine can be run up to 1500 r.p.m., although a speed of 1000 is sufficient for this job. With the $2^1/_4$-in. cutters shown this gives a surface speed of 580 ft. per min. The hand feed can be operated about as fast as the operator can push the lever and the time required is about 3 sec. per tooth. These pinions have 12 teeth, and three are finished at one setting, the indexing and operation of the locking pin being automatic. This together with the loading time of 30 sec., gives a total of 66 sec., or a total time of 22 sec. per gear.

While these examples are limited largely to a particular line of machines, they are nevertheless typical of the problems which may arise and the methods of handling work. And it is believed that enough of them have been given to indicate that special study of each particular job will point the way to improved time-saving methods and lead eventually to the adoption of the best.

The Railway Centenary Celebrations

OF all the steps into which the progress of development of mankind from savagery through barbarism to civilization may be divided up, the greatest in influence and results was made when man first learned to generate and control power. In entering on this phase of progress, the engineers of old started a movement without the effective progress of which the people of the world would have still remained in a state of primitive barbarism. The pioneer engineers and those who make the fundamental discoveries must therefore be regarded as among the finest types of humankind and as worthy of the highest distinctions bestowable by their fellow men, though, it must be admitted that recognition is sometimes given tardily, if at all. One hundred years ago, George Stephenson, by the exercise of a mentality rich in inventive capacity, was enabled to start a movement, the consequences of which to the world, despite his foresight, he could never have estimated. We, in this centenary year, are able to look back and gage the significance of his work, and the many steps in the progress of its development up to the present time. The gains have been not merely material, for from the start of railroad transport we must date the improved relations between peoples of different race and their better appreciation of each other's nature and culture. It was therefore fitting that testimony to the value of Stephenson's work should be made this year to the entire world, and our railway engineers are to be commended for their inspiration in inviting the Railway Congress to hold its tenth session in this country at a time when estimate could well be made of what we owe to a pioneer engineer and an illustrious Englishman. Happier still was the decision to put the old locomotives made by Stephenson back again on their historic line and show them running in association with British engines and rolling-stock of every period up to the present. That the pageant was an exceptional success will be acknowledged by all who were privileged to witness it, thanks to the hospitality of the London and North Eastern Railway Company, which dates its origin from the foundation of the original Stockton and Darlington line. Organization could be carried to no higher level than was shown in this demonstration of the working of Stephenson's engines and their relationship to modern designs.

The celebrations were begun at Darlington on Wednesday, July 1, when H.R.H. the Duke of York opened an exhibition of railway relics at the Faverdale Works of the London and North Eastern Railway Company, to which all the British railway companies, many museums and individuals, had lent their historic treasures. In this collection, the student of the history of the development of railways was provided with the possibility of inspecting the original drawings of the pioneers and of studying the construction of the actual components of railway appliances of every age, as well as of contrasting the actual locomotives and rolling-stock of different periods in the history of British railways. It was from these engines, carriages and wagons that the 53 representative examples were taken along the line of the Stockton and Darlington Railway in a procession before the Duke and Duchess of York and the assembled railway authorities of our own and other countries. In the parade, first place was given to an engine built by George Stephenson and Nicholas Wood in 1822 at the Hetton Colliery workshops, prior to the establishment of Stephenson's original works at Newcastle-on-Tyne.

A banquet was held on Thursday evening, after the parade of locomotives, at which Viscount Grey of Fallodon presided over a company of 1000 guests. In speaking on "The Centenary of British Railways," he traced what the world of today owed to the pioneer work of George Stephenson and his associates in the development of railways. These men were not speculators, he said, but had the driving force in their minds that they were doing something to develop the resources of the country. In replying, Sir Arthur Pease, Bart., spoke of the present conditions of the railways and the necessity for united sacrifice on the part of the whole community to meet the circumstances in which the trade of the country had now to be conducted. The delegates to the Railway Congress were also entertained at Darlington by the London and North Eastern Railway Co., when they visited the exhibition on Friday, July 3. They were received by William Whitelaw, the Chairman of the company, who after luncheon delivered an address of welcome to the Congress. Later a presentation was made by Cavalliero Chiarugi of a bronze shield on behalf of the Italian railwaymen, to the British railwaymen, and Dr. Wang presented a Chinese scroll. (*Engineering*, July 10, 1925, p. 41.)

A Code of Design for Mechanical Springs

By JOSEPH KAYE WOOD,[1] NEW YORK, N. Y.

This paper gives a new derivation of the general spring formulas developed by the author from the numerous existing orthodox spring formulas and presented in his paper on Mechanical Springs at the A.S.M.E. Annual Meeting in December, 1924. The two sets of formulas, although derived independently, are identical in form. In view of the general manner in which the new derivation establishes the logical arrangement of the formulas, the author has drawn up and included in the paper a brief code of design for mechanical springs, which he hopes will serve as a nucleus for a more complete code in the future.

AT THE Annual Meeting of The American Society of Mechanical Engineers in December, 1924, the author presented a paper[2] on mechanical springs in which a general method of design for springs was described. Since that time further study of the general method has led him to the discovery of a general derivation for all spring formulas, which makes the method so logical as to justify its use as a basis for codification.

The development of such a code, although tentative and unofficial, can be made to serve two useful purposes; namely, to provide a clearer vision as to the ultimate or final code desired, and in the meantime to enable the design of mechanical springs to be accomplished in a more efficient manner than possible in the present haphazard state of the art.

The paper referred to gave the derivation of the following formulas, which, while in accordance with admitted principles of elasticity, are expressed in such a form as to be applicable to mechanical springs in general.

Load-Deflection Rate:

$$\frac{P}{F} = kE\,\frac{A}{l}\left(\frac{d}{D}\right)^2 \qquad [1]$$

Safe Maximum Load:

$$P_m = mSA\left(\frac{d}{D}\right) \qquad [2]$$

Safe Maximum Deflection:

$$F_m = n\,\frac{S}{E}\,l\left(\frac{D}{d}\right) \qquad [3]$$

Safe Maximum Work:

$$W_m = \frac{u}{2}\,\frac{S^2}{E}\,Al \qquad [4]$$

New Derivations of General Spring Formulas

Let it be assumed for the moment that only one method of stressing a bar is known, namely, that of pure tension. The bar shown in Fig. 1, l in. long and of uniform cross-sectional area A sq. in., is stretched F_t in. by a force of P_t lb. The resulting unit stress S in lb. per sq. in. is equal to P_t/A, while the load-deflection rate P_t/F_t in lb. per in. is evaluated as follows:

$$\frac{P_t}{F_t} = E\,\frac{A}{l}$$

where A/l is the bar ratio and E is the tensile modulus of elasticity.

In the above equation the bar ratio might be considered as an operator which modifies the impractically high value of E (the lowest value of E for metals is equal to 6,000,000) so as to give a much lower load-deflection rate. Suppose, for example, a rate of 1000 lb. per in. is desired for a metal having a value of E of about 6,000,000 lb. per sq. in. The bar ratio would need to be equal to 1/6000, and if a maximum deflection of 1 in. is specified, the area component of this ratio for a given elastic limit of 30,000 lb. per sq. in. would need to be 0.033 sq. in. This would limit the length l in the bar ratio to a minimum of 200 in., which in a straight tensile piece would be impractical for designing purposes, as space limitation is of importance. Reducing the length to the desired value increases the load-deflection rate to a prohibitive amount. In other words, with the resilient energy obtained from a reasonably short length of bar having a moderately small cross-sectional area, the load component is much too large and the deflection component much too small to give a load-deflection rate of a value suitable for use in mechanical design. The question arises, therefore, that with the limitations imposed by the requirements of mechanical design, how can the deflection component be increased relatively to its load component, or in other words, how can the load-deflection rate be decreased?

Since the maximum resilient energy is

$$W_m = \frac{S^2}{2E}\,Al$$

and since for a given material the volume Al cannot be varied sufficiently to give the desired result, some external means must be employed. Fig. 2 shows an external device in the form of a mechanical lever whose arms are d and D. By applying a force P to the end of arm D in the direction shown and moving it through a distance F, the bar of length l and cross-sectional area A is stretched an amount equal to F_t. Then

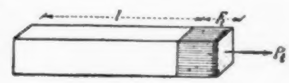


Fig. 1

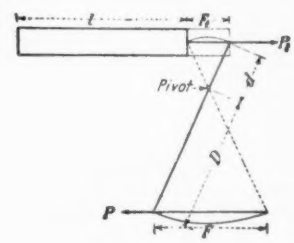


Fig. 2

$$P_t = \frac{D}{d}P \quad \text{and} \quad F_t = \frac{d}{D}F$$

Combining,

$$\frac{P_t}{F_t} = \left(\frac{D}{d}\right)^2\frac{P}{F}$$

But we know that

$$\frac{P_t}{F_t} = E\,\frac{A}{l}$$

hence

$$\frac{P}{F} = E\,\frac{A}{l}\left(\frac{d}{D}\right)^2 \qquad [5]$$

in which the square of the lever index appears as in Equation [1].

Proceeding in the same manner,

$$P_t = \frac{D}{d}P$$

in which $P_t = SA$, giving

$$P = SA\left(\frac{d}{D}\right) \qquad [6]$$

and in this expression appears the first power of the lever index as in Equation [2].

From [5] and [6]

$$F = \frac{S}{E}\,l\left(\frac{D}{d}\right) \qquad [7]$$

and evaluating $W = \dfrac{PF}{2}$ the following expression for work is obtained:

$$W = {}^1/_2\,\frac{S^2}{E}\,Al \qquad [8]$$

[1] Consulting Engineer, Assoc-Mem. A.S.M.E.

[2] See Mechanical Engineering, April, 1925, vol. 47, no. 4, p. 258.

Contributed by the Machine Shop Practice Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers. Abridged.

which of course is the same whether a lever is used or not used.

It will be noted that Formulas [5], [6], [7], and [8], derived in a new and different manner, are similar to the generalized orthodox Formulas [1], [2], [3], and [4], with the exception that the constants k, m, n, and u, respectively, are missing. These constants can be supplied by considering that the minor arm of the lever index is some fractional part of d, which is not identified yet as the depth or diameter of the bar's section. This modification would furnish constants for all formulas except [8], since the resilient energy obtained is independent of the lever system. A constant, less than unity, can be inserted in the latter formula by introducing the idea of stress efficiency, which is assuming that all fibers in the bar are not stretched to the maximum value. As the minor lever arm moves through the distance corresponding to the maximum stretch, even though only a very small portion of the fiber bundle is stretched to

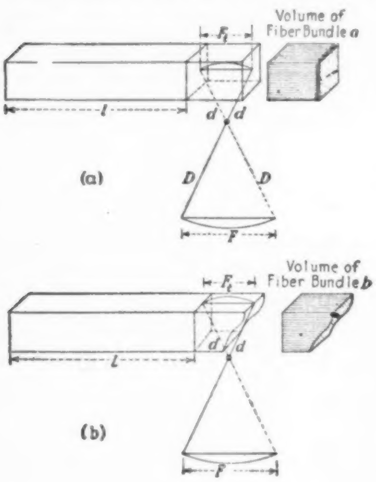


Fig. 3

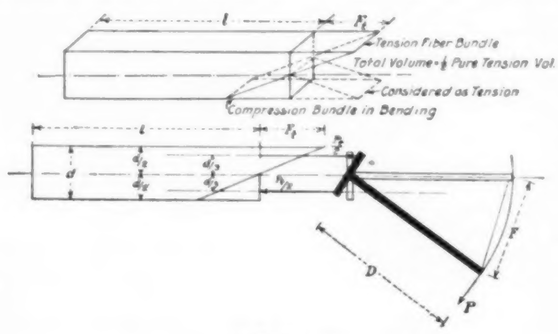


Fig. 4

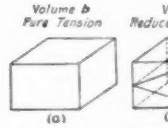


Fig. 5

this amount, the load P is decreased in proportion. This constant multiplied by 100 might well be called the "per cent efficiency in stressing," and since it affects the load component of resilient energy, it accordingly affects the load-deflection rate directly. Fig. 3(a) shows a 100 per cent efficient fiber bundle and Fig. 3(b) shows one of reduced efficiency. The ratio of the volume of b to that of a multiplied by 100 is the per cent efficiency of stressing in b. Although the length of travel is the same for both lever systems, the load is less in b because less resistance has to be overcome.

Assigning constants, therefore, to Formulas [5], [6], [7], and [8] in accordance with the above reasoning and using for these constants the same letters as were used for Formulas [1], [2], [3], and [4], namely, k, m, n, and u, a set of formulas is obtained that is identical with the latter when viewed from a general standpoint. Thus the same general spring formulas are obtained directly, by what may be considered as the lever principle, as were obtained by generalizing many individual formulas, derived separately by long and somewhat complicated methods.

Evaluation of Constants

To evaluate the constants k, m, n, and u requires more detailed consideration of the fiber bundle for the various methods of stressing a bar. Take, for example, the type of stressing shown in Fig. 4, in which the pivot of the lever system is placed on the geometric axis of the bar and the minor arm consists of two right-angled extensions. One-half of the stressing is compression, but the correction due to the difference between the tensile and compressive properties of the same material is a refinement hardly necessary at this time, although it would be worthy of a separate consideration. With this qualification, therefore, the value of u might be determined by comparing the volumes shown in Fig. 5. Volume a is equal to one-half of volume b, which indicates that the efficiency in stressing is 50 per cent. To determine k, it must be remembered that this constant involves two factors, one due to a modification of the minor lever arm d and the other due to the reduced efficiency which affects only the load component. If now it is specified that the section of the bar is square, and that the vertical dimension (depth) is d, then the modification factor is $(^1/_3)^2$ since the resultant load on the fiber bundle is concentrated at the center of gravity. The load component is decreased by one-half, as is the efficiency, because the loss in the latter is all taken in the former. Multiplying these two factors would give a value of $^1/_{18}$ for k. Similar reasoning would indicate a value of $^1/_3 \times {}^1/_2 = {}^1/_6$ for the value of m and 3.00 for the value of n.

Comparing these values with those of uniform bending, as in helical and spiral springs stressed flexurally, which this type of stressing would seem to approximate, the following results are obtained.

Constant	k	m	n	u
In old derivation	1/12	1/6	2	0.33
In new derivation	1/18	1/6	3	0.50

Although these comparisons show appreciable differences, they are not very great considering that the derivations were made by two absolutely independent methods.

Fig. 6 illustrates the relative values of the constants k, m, n, and u for several methods of stressing. The basis of comparison is that the bar dimensions, maximum stress, and lever index are the same in all cases. Thus in the tensile triangle of resilient energy the load and deflection legs are both equal to unity, while in the torsional triangle the deflection leg is greatly increased at the cost of 50 per cent efficiency in

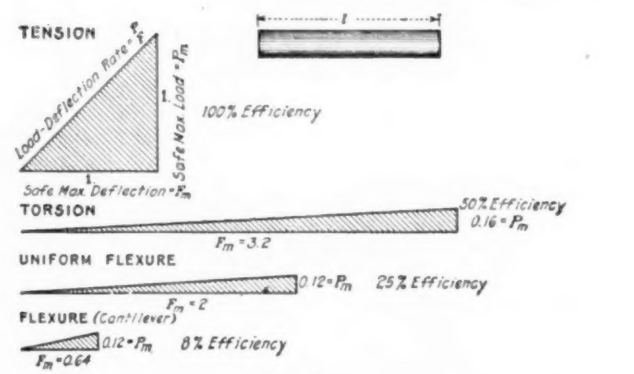


Fig. 6

stressing. In the energy triangle for cantilever flexure the efficiency is only 8 per cent, and nothing is gained for the sacrifice of 92 per cent efficiency since the deflection and load legs are both less than unity. Uniform flexure is therefore preferable to the latter type of flexure when considered on this basis.

With the foregoing principles established on a logical and sound basis, the author feels that the following tentative code, which is constructed in accordance with these principles, should prove useful and valuable. Until research has either confirmed or disproved the new values of the constants the values derived from the orthodox formulas will be used therein. It should be remembered, however, that the new derivation of the generalized formulas is fully applicable in any case up to the point of valuation of constants for specific cases.

TENTATIVE CODE OF DESIGN FOR MECHANICAL SPRINGS

General Definitions

A mechanical spring is an elastic body whose load-deflection rate and maximum safe deflection are of values suitable for mechanical use.

A mechanical-spring material is an elastic material of the kind which, when made into bodies of a shape and size suitable for use in mechanical design, will function repeatedly and permanently as a mechanical spring.

Mechanical Service Requirements

A certain point in a mechanical member having practically a

rigid construction shall require contact with the load or free end of a spring in such a manner as to exert a pull or pressure of P_0 lb. After a travel of F_1 in., a pull or pressure of P_1 lb. shall be exerted. The load-deflection rate required of the spring shall therefore be equal to $\frac{P_1 - P_0}{F_1}$ as illustrated in Fig. 7.

The motion of the point of contact shall be either translatory or

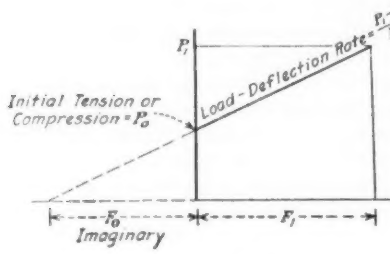


FIG. 7

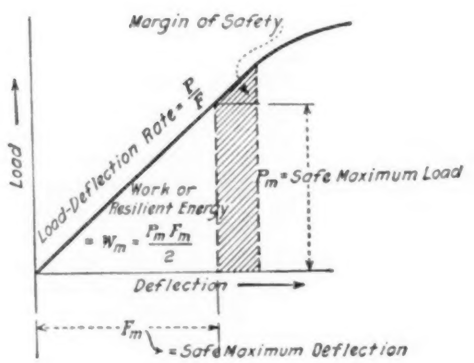


FIG. 9

rotatory and shall fall within a satisfactory range of periodicity. Types of springs giving translatory motion are helical, stressed torsionally, and flat, stressed flexurally, while springs giving rotary motion are those of the helical and spiral types stressed flexurally.

The spring selected shall perform the required service within the space assigned. The space-limitation rectangle and the load line or path of motion of contact point are fixed for specific service requirements. Various cases which might arise are shown in Fig. 8.

FORMULAS[1]

Required	Constant	Material	Bar	Lever ratio

Load-deflection rate:

$$\frac{P}{F} = k \times E \times \frac{A}{l} \times \left(\frac{d}{D}\right)^2 \quad \text{[I]}$$

Safe maximum load:

$$P_m = m \times S \times A \times \left(\frac{d}{D}\right) \quad \text{[II]}$$

Safe maximum deflection:

$$F_m = n \times \frac{S}{E} \times l \times \left(\frac{D}{d}\right) \quad \text{[III]}$$

Safe maximum work:

$$W_m = \frac{u}{2} \times \frac{S^2}{E} \times Al \quad \text{[IV]}$$

Other formulas are—

Total weight of spring material:

$$w = (Al \times \delta) + \text{elastically dead material} \quad \text{[V]}$$

Weight per linear foot:

$$\frac{w}{l} = (A \times \delta) + \text{elastically dead material} \quad \text{[VI]}$$

[1] Formulas [I]–[IV] are the same as Formulas [1]–[4].

Period of oscillation:

$$T = \frac{2\pi}{\sqrt{g}}\sqrt{\left(K^2Q + \frac{w}{i}\right)\frac{F}{P}} \quad \text{[VII]}$$

The above formulas can be used to calculate any type of mechanical spring to meet a given service requirement when the terms are fully understood and the values of the constants are known. These terms are to be used as follows:

SERVICE REQUIREMENTS

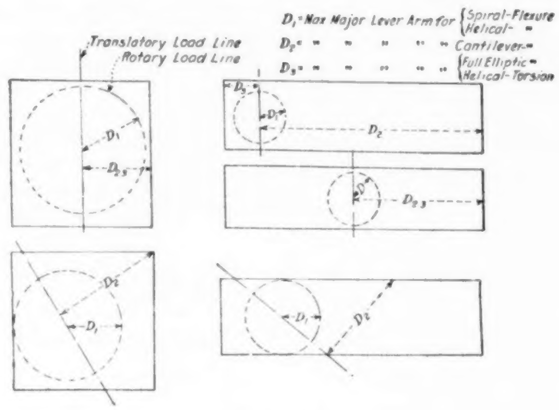


FIG. 8

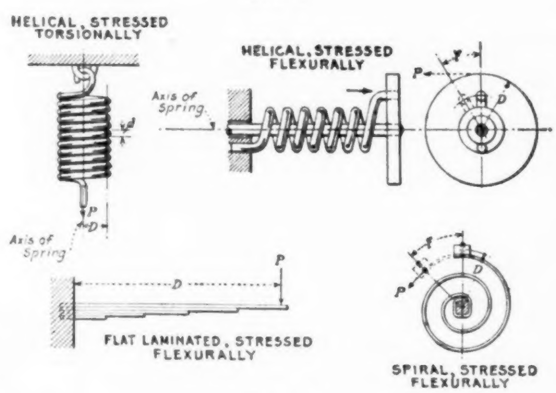


FIG. 10

The diagram in Fig. 9 shows the significance of the load-deflection rate P/F, safe maximum load P_m, safe maximum deflection F_m, and the safe maximum work W_m. The units used are pounds and inches, an inch-pound being the measure of work or resilient energy.

The total weight of spring material w plus elastically dead material is in pounds. By "elastically dead material" is meant that material which does not furnish elastic energy, such as the clamped portion of automotive leaf springs, the squared or looped ends of helical springs, and also the slight wastage involved in manufacture. "Total weight per linear foot" means the weight of a section of bar one foot long, assuming the cross-section to be constant. The latter figure is useful in computing the total cost of the spring bar or plate required.

The period T is the number of seconds required for one complete oscillation of the load end of the spring while either constantly carrying a weight of Q lb. or just its own weight w lb., i.e., vibrating freely.[1]

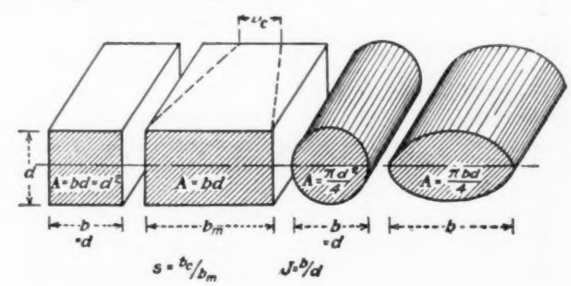


FIG. 11

LEVER RATIO

The lever ratio or index D/d consists of the minor arm d and major arm D, both in inches. For simplicity, the minor arm is made equal to the depth or diameter of the bar or plate section, but actually this arm varies directly from zero to $d/2$, the resultant value being $d/3$. The major arm D is in general equal to the distance from the load axis, whether it be a straight line or a circle, to the clamped end or geometrical axis of the spring. Fig. 10 gives four of the principal types of springs showing the major arm D. In the case of the hour-glass, barrel, and conical types of helical springs, whether stressed torsionally or flexurally, the mean radius of the maximum-sized coil is taken as D and a factor is included in each of the constants k, m, and n allowing for variation in the sizes of the other coils.

[1] This general periodicity law for mechanical springs was derived by the author in 1922 and published in the *American Machinist*.

BAR

The cross-sectional area of the bar, equal to A sq. in., may have any of the shapes shown in Fig. 11, the dimensions being as shown. In the case of rectangular and elliptical sections stressed in torsion, a factor J equal to the ratio b/d is included in the constants k, m, n, and u. Likewise, in the case of flat laminated springs that have trapezoidal plan developments, a factor s, equal to the ratio b_e/b_m, is included in the same constants. The minimum width at the free-end section is b_e and the width of the maximum section at the clamped end is b_m. The total length of the bar or plate, excluding the elastically dead portion, is l in. In flat laminated springs this length l is obviously equal to the major arm D of the lever ratio, while in other types of springs, such as the helically or spirally wound types, the relation of l to the mean radius of the coil R and the number of coils or convolutions is determined by ordinary geometry.

For the helix,

$$l = 2\pi RN$$

where N equals the number of coils. In the case of the torsionally stressed type, l is equal to $2\pi DN$.

For the spiral,

$$l = 2\pi \frac{(R + r)}{2} N \text{ (approximately)}$$

where R is the radius of the outside convolution and r that of the inside convolution, while N equals the number of convolutions. The greater the number of convolutions, the greater will be the accuracy of the above approximation.

Al is the volume of the active material in the bar in cubic inches, while $\frac{A}{l}$ is the bar ratio.

TABLE 1 STRESS-METHOD AND FORM-OF-SECTION CONSTANTS

Spring Type of	Method of Stressing	Form of Section	k	m	n	u	Per Cent Efficiency in Stressing
Helical	Torsion	Round	0.05	0.16	3.20	0.50	50
"	"	Square	0.08	0.19	2.25	0.42	42
"	"	Elliptical	$0.10\left(\frac{J^3}{1+J^2}\right)$	$0.22\left(\frac{J^2}{\sqrt{1+J^2}}\right)$	$2.3\left(\frac{\sqrt{1+J^2}}{J}\right)$	$0.50J$	$50J$
"	"	Rectangular	$0.16\left(\frac{J^3}{1+J^2}\right)$	$0.27\left(\frac{J^2}{\sqrt{1+J^2}}\right)$	$1.6\left(\frac{\sqrt{1+J^2}}{J}\right)$	$0.42J$	$42J$
Conical	"	Round	0.10	0.16	1.60	0.25	25
"	"	Square	0.16	0.19	1.12	0.21	21
Helical	Flexure	Round or Elliptical	0.06	0.12	2.0	0.25	25
"	"	Square or Rectangular	0.08	0.16	2.0	0.33	33
Spiral	"	Round or Elliptical	0.06	0.12	2.0	0.25	25
"	"	Square or Rectangular	0.08	0.16	2.0	0.33	33
Cantilever	"	Round or Elliptical	0.19	0.12	0.65	0.08	8
"	"	Square or Rectangular	0.25	0.16	0.65	0.10	10
Cantilever, Triangular Plan [1]	"	Rectangular	0.16	0.16	1.00	0.16	16
Cantilever, Trapezoidal Plan [1]	"	"	$1.00\left(\frac{s+2}{12}\right)$	0.16	$\frac{2}{S+2}$	$\frac{0.32}{S+2}$	$\frac{32}{S+2}$

Note: $s = \frac{b_e}{b_m}$ (see Fig. 11)

$J = \frac{b}{d}$ (see Fig. 11)

Assumed: $G = 0.4E$ and $S = 0.63S$ for tension

[1] Basic for laminated springs.

MATERIAL

The tensile or Young's modulus of elasticity in lb. per sq. in. is designated by E. The torsional modulus of elasticity G in lb. per sq. in. is usually equal to about two-fifths of E, and the values of the constants k, n, and u in Table 1 are based upon this factor. Values of E given in Table 2 for different metals show that they range from 6,000,000 to 60,000,000 lb. per sq. in. and that they tend to vary in accordance with the corresponding densities δ. The latter figure is used in connection with the calculation of weights of spring material required. Very few conditions appreciably affect the value of Young's modulus for a given material, and even these to only a slight extent.

Probably the greatest variation in the modulus for a metal of a given composition is produced by variations in the surrounding temperature. Thus, the modulus for steel is reduced about 10 per cent by elevating the surrounding temperature from about 70 deg. fahr. to 300 deg. fahr. This fact should be taken into consideration, especially when designing springs for steam and gasoline apparatus such as turbines and reciprocating engines.

The maximum safe working stress is usually based upon the proportional limit S in spring design, and for the present the safe stress will be considered equal to this limit, after which the manner of allowing for the unknown factors such as fatigue, impact, etc. will be explained.

The value of S varies over a very large range due to many conditions, such as slight variations in composition, heat treatment, cold working, surrounding temperatures, etc. Greater care should therefore be exercised in ascertaining the correct value of S than in finding the value for E because the value of the modulus E varies but slightly under these different conditions. The value of S in torsion is generally equal to about 0.63 of the value of S in flexure or tension, and this factor is included in constants m, n, and u for springs stressed torsionally. In any case, for an accurate comparison of different metals the values of S and E should not be used unless they actually represent the metals under consideration. The table, which gives these values for tensile stressing along with the corresponding material index and other properties, should be used only for general purposes, as the figures were collected from many diverse sources.

The working stress S has been assumed equal to the proportional limit, which does not provide any margin of safety against permanent deformation and fatigue failure. In view of this, the author has suggested that the material index S/E be divided into three types: the *static* index, to be used in spring design where the number of deflections is less than one million; the *kinetic* index, to be used where the number of deflections is considerably greater than one million; and the *elevated-temperature* index, to be used at temperatures above 500 deg. fahr. Table 2 shows the value of the static material index for each of the metals listed and of the kinetic material index for those metals whose endurance limits have been determined. The modulus is the same for both of these indices, because fatigue apparently does not affect this property.

Both the modulus and the proportional limit are decreased appreciably when the surrounding temperature is increased, but the proportional limit usually decreases to a much greater extent, giving rise to a decrease in the elevated-temperature index. The character of this variation would be different for different metals, which necessitates a special determination of both S and E at the temperature desired, in order to compute the material index. Indices for both monel metal and high nickel-chrome steels at 1000 deg. fahr. are probably not less than 60 per cent of the indices at room temperature, as compared with about 40 per cent for some ordinary carbon steels. Springs required to operate at high temperatures should be made of metals whose material indices will not decrease too much.

In actual spring design there are additional factors which determine the maximum safe working stress, such as occasional overload, sudden loads, and impact loads, but which need not be

considered in the comparison of two metals unless the material indices are somewhere nearly equal in value.

In accordance with the above principles the static, kinetic, or elevated-temperature material indices, as the case may be, should be modified by a factor of safety giving what might be called the "safe working material index" equal to S/E for use in Formula [III]. S and S^2/E should be treated similarly before using in connection with Formulas [II] and [IV], respectively.

S^2/E is the modulus of resilience in inch-pounds per cubic inch. Since S is squared in this term, and since it is subject to greater variation than E, the variation of this modulus will be much greater than in the case of the material index.

Service requirements sometimes specify freedom from corrosion due to various influences. Atmospheric corrosion would prohibit in certain cases the use of the best kind of spring metal, namely, steel. For small steel springs it is possible, although difficult, to protect against such corrosion by plating the steel with tin, nickel or a combination of nickel-copper-nickel, while for medium-sized springs protection for a short time may be had by painting. This problem is solved more satisfactorily in cases where space limitations permit by resorting to the use of non-rusting steels, phosphor bronze, nickel silver, monel metal, nickel, and brass. The last-named metal is subject to season cracking, however, particularly if under sustained stress.

Steel as a spring material is prohibited also where a spring must be non-magnetic, except in the case of normally austenitic steel, which, however, has an inferior material index and modulus of resilience.

Constants k, m, n, u

These constants vary with method and efficiency of stressing, form of bar or plate cross-section, and with non-uniformity in size of bar or plate cross-section. The values of these constants for some of the most important types of springs are given in Table 1. As already stated, these constants may contain either a factor s or J, or both. Factor s is the ratio of the minimum width b_e of bar or plate having a trapezoidal form and of constant thickness d to the maximum width b_m. The area A used in the formulas in this case is $b_m d$. Factor J is the ratio of the breadth b to the depth d of rectangular and elliptical sections of bars stressed torsionally.

Special Terms and Constants

Q is the constant weight in pounds that an oscillating spring carries, such as the sprung weight of an automobile or locomotive. If the spring vibrates freely Q equals zero and hence vanishes along with K^2. Where Q is large compared with w or the weight of the spring, w/i practically vanishes.

The gyration factor K is equal to unity in all cases except that of the balance wheel on either spiral or coiled springs, that is, where the weight Q gyrates about the axis of the spring. In these cases K is equal to the radius of gyration divided by D. This is a particular case of a circular load line referred to in a paragraph on Service Requirements.

The term i is called the "equivalent mass factor," and its value depends upon the mode factor V. Values of i for the various types of springs vibrating freely in the fundamental and higher modes are given in Table 3.

The constant g is the acceleration of gravity, which is taken as 386 in. per sec. per sec.

Brief Summary of Terms and Constants in Formulas

$\dfrac{P}{F}$ = load-deflection rate, lb. per sq. in.

P_m = load, safe maximum, lb.

F_m = deflection, safe maximum, in.

W_m = work, safe maximum, in-lb.

$\dfrac{S^2}{E}$ = modulus of resilience, in-lb. per cu. in.

$\dfrac{S}{E}$ = material index, safe working (pure number)

S = unit tensile stress, safe maximum, lb. per sq. in. (= $1.6S$ for torsion)

E = tensile modulus of elasticity, lb. per in. (= $2.5G$, where G = modulus for torsion)

Al = volume of bar, cu. in.

$\dfrac{A}{l}$ = bar ratio, in.

A = area of bar cross-section, sq. in.

l = length of bar, in.

$\left(\dfrac{D}{d}\right)$ = lever index (pure number)

D = major lever arm or moment arm of load, in.

d = minor lever arm (twice maximum) or depth of section, in.

k = load-deflection-rate constant

m = safe-maximum-load constant

n = safe-maximum-deflection constant

TABLE 2 VALUES OF MATERIAL INDEX AND OTHER PROPERTIES FOR VARIOUS METALS

Reference Number	Material	Form of Material	Density in Lb. per Cu. In. = δ	Young's Modulus in Lb. per Sq. In. = E	Proportional Limit in Lb. per Sq. In.	Material Index (Static)	Endurance Limit in Lb. per Sq. In.	Material Index (Kinetic)	Tensile Strength in Lb. per Sq. In.	Ult. Elongation, Per Cent in 2 in.	Percentage Reduction in Area
1	Magnesium alloy (Zn—4%)	Bar	0.065	6,000,000	6,000	0.0010	.....	.....	36,000	17	20
2	Duralumin	Bar	0.100	10,000,000	25,000	0.0025	.....	.....	50,000	16	50
3	High-nickel-chrome steel	Rolled sheet	0.218	27,000,000	100,000	0.0037	.....	.....	127,000	50	55
4	High-nickel-chrome steel	Cold-drawn wire	0.220	28,000,000	180,000	0.0064	.....	.....	220,000	12	..
5	Chromium steel	Rolled sheet	0.278	29,000,000	150,000	0.0052	.....	.....	185,000	15	35
6	Low-nickel-chrome steel	Bar	0.280	30,000,000	115,000	0.0038	.....	.....	138,000	18	62
7	Spring carbon steel (1.0% C.)	Rolled sheet	0.280	30,000,000	152,000	0.0056	.....	.....	168,000	11	..
8	Carbon steel (1.20% C.)	Bar	0.280	30,000,000	120,000	0.0040	92,000	0.0036	180,000	9	15
9	Steel music wire	Drawn wire	0.285	30,800,000	260,000	0.0084	.....	.....	310,000	8	..
10	Silicon-manganese steel	Bar	0.282	30,000,000	100,000	0.0033	65,000	0.0022	157,000	16	40
11	Spring molybdenum steel	Sheet	....	30,000,000	215,000	0.0071	.....	.....	225,000	14	40
12	Nickel steel	Bar	0.280	30,000,000	82,000	0.0027	63,000	0.0021	112,000	24	60
13	Pure tungsten	Small drawn wire	0.720	60,000,000	370,000	0.0061	.....	.....	490,000	..	..
14	Brass (Zn—35%)	Rolled sheet	0 307	12,000,000	40,000	0.0033	.....	.....	60,000	25	30
15	Phosphor bronze (8% Tin)	Rolled sheet	0.312	16,000,000	70,000	0.0044	40,000	0.0025	90,000	9	..
16	Nickel silver (Ni—18%, Zn—12%)	Rolled sheet	0.320	17,000,000	65,000	0.0038	.....	.....	90,000	3	33
17	Monel metal	Rolled sheet	0.320	23,000,000	50,000	0.0022	.....	.....	94,000	30	35

TABLE 3 VALUES OF i FOR DIFFERENT FORMS OF SPRINGS

Directions of Oscillations		Form of Spring	Condition of Ends F = Fixed R = Free	Value of v for the Fundamental Mode	Values of v for the Higher Modes	Values of i For Any Of the Modal Values m	Values of i With Light or Without Any Attached Weight v = Fundamental
Longitudinal		Helical	1F, 1R	1	3, 5, 7, 9, etc.	$4 \div \pi^2 v^2$	0.40
		Helical	2F or 2R	2	4, 6, 8, 10, etc.	$4 \div \pi^2 v^2$	0.40
Transverse		Cantilever	1F, 1R	1.194	3, 5, 7, 9, etc.	$48 \div \pi^4 v^4$	0.25
		Buffer	2 supported	2	4, 6, 8, 10, etc.	$768 \div \pi^4 v^4$	0.50
		Buffer	2 free				
			Consider one overhanging end	1.194	3, 5, 7, 9, etc.	$48 \div \pi^4 v^4$	0.25
		Spiral	1F, 1R	1.194	3, 5, 7, 9, etc.		
		Coiled	1F, 1R	1.194	3, 5, 7, 9, etc.		

u = safe-maximum-work constant
w = total weight of spring material, including elastically dead material, lb.
δ = density of spring material, lb. per cu. in.
T = period of oscillation in seconds
g = acceleration of gravity = 386 in. per sec. per sec.
Q = weight of mass attached to oscillating spring, lb.
K = gyration factor, equal to the radius of gyration divided by D (pure number)
i = equivalent mass factor depending on mode of oscillation (pure number).

General Procedure in Design of Spring

Referring to Fig. 8,

1 Determine maximum area and form of space allowed for spring in machine or mechanism.

2 Determine location, direction, and shape of load path required.

3 From operation requirements of machine or mechanism determine load-deflection rate, maximum travel, or load, and if important, the period of oscillation.

4 Select types of springs that fulfil 1 and 2 and then for each type calculate the spring dimensions, using Formulas [I] and [III] (or [II]), to fulfil 3. Using Formula [IV], check the results and select the best spring of the various types calculated.

5 The following general precautions should be observed.

a The material constants E and S for the proposed spring material should be known as accurately as possible

b The lever index D/d for helical springs should not be less than 1.5 in any case, and preferably not less than 3. Between 1.5 and 3 the value of the lever index is permissible only when unavoidable, because of space limitations and other causes

c The manner of securing or clamping the inactive part of springs should be considered very carefully in order to eliminate high local stresses. This is likely to occur in the end loops or squared ends of helical springs and at the bands of elliptic springs

d The unequal distribution of stress in laminated springs, due to deviation of the actual developed plan view from the theoretical plan view and to the practice of "nipping" during the process of manufacture should be given very careful attention. The same precaution should likewise be taken in the case of high localized stresses due to nipping, clipping, and the practice of putting center pins through spring blades.

Conclusion

In the foregoing exposition of the author's general method of spring design in brief codified form, preceded by a new and simple derivation of the general formulas that emphasizes the logic and simplicity of the method, it is felt that an initial start has been made toward the formulation of a permanent code of design for mechanical springs. No attempt has therefore been made to present a code complete in every detail.

Discussion

IN THE discussion which followed the presentation of J. K. Wood's paper on the design of springs the following participated: T. McLean Jasper,[1] W. G. Brombacher,[2] and Dayton A. Gurney.[3] The former two had presented written discussions.

Mr. Jasper suggested that Formula [4] for the safe maximum work be generalized to include all kinds of elastic bodies.

Mr. Brombacher agreed with the author that the old value for the constant k in the load-deflection formula was too high, but not quite as low as proposed by the author, as certain experiments with a particular group of phosphor-bronze helical springs would tend to indicate. It was significant, however, that the discrepancies should be in the same direction as indicated in Mr. Wood's paper. Attention should also be called to the fact that the values of the stress-method and form-of-section-constants given in Table 1 were subject to modification in the cases where the method of stressing was torsional, owing to the fact that the modulus in torsion had been assumed to be $^2/_5$ of the value of Young's modulus E. Deviations from this assumption existed which were of practical importance.

Mr. Brombacher noticed that certain factors in the performance of springs were not mentioned in the code which were, however, of particular importance in precision-instrument design. He suggested the inclusion into the code of the following definition:

"*Hysteresis* is the difference in the deflection of a spring for any given load in the load-deflection cycle when the load is first increased from an initial value to a maximum value and is then decreased to the original value. It is desirable that the hysteresis of a spring be a minimum in any case where the reading of an instrument depends on the deflection of the spring."

Mr. Gurney complimented the author on his paper and said that the Ordnance Department as a large consumer of springs was very much interested in the development of a standard method of design. The requirements of springs, particularly for counter-recoil and equilibrator springs, were quite severe, and it was important that the loads and deflections should be held within narrow limits.

The author, in closing, said that the expansion of Formula [4] to that given by Professor Jasper could have readily been included in the code, but after consideration of such a formula when preparing the paper he had decided that since Poisson's ratio did not vary widely for any one material, particularly steel, the simpler relations $G = {}^2/_5\, E$ and S in torsion = $0.63\,S$ in tension were sufficiently accurate for practical purposes. The paper specifically stated that in cases where a high order of accuracy was necessary the exact relation between these various properties should be determined experimentally. Long, unnecessary mathematical expressions would only tend to defeat the purpose of the paper, namely, simplicity or logical arrangement. Proof of the above relations between the tensile modulus and the torsional modulus on the one hand, and between the tensile working stress and the torsional working stress on the other, was as follows:

It was safe to say that no matter what type of stress constants, tensile or torsional, were considered, only a definite amount of energy could be obtained from a bar of material. If this was true, then the conversion of S^2/E for tension into S^2/E for torsion in the energy Formula [4] should not introduce any factor different from unity. Thus the square of 0.63 was 0.3969, which divided by $^2/_5$ equaled 0.992, which was close to unity.

The author did not agree with Professor Jasper that the constant u should be changed to suit dynamical constitutions unless the form of the fiber bundle was altered. Instead the value of S should be changed accordingly.

W_m was not controlled by the elastic limits. In the case of the "kinetics" condition the endurance limit would be the controlling factor.

It was gratifying to learn that the experimental results obtained by Mr. Brombacher had furnished further confirmation of the fact that the old value of the constant k was too high. The only reason that could be advanced at the present time for the author's value of k being lower than that of Mr. Brombacher's was that the former's value corresponded to a maximum deflection, when the error was greatest, while the deflections employed by the latter were probably not so extreme.

As regarded Mr. Brombacher's remark on the use of the assumption that the modulus in torsion was equal to $^2/_5$ of the value of Young's modulus E, the author had already stated his reasons for using this factor and its limitations in his reply to Professor Jasper.

The author, although fully realizing the importance of hysteresis and after-effect in precision-instrument design, felt that these considerations were details which could be allotted to a separate section in a more complete and expanded code, using the present code as a basis. He had touched upon these factors in his previous paper on mechanical springs, referred to at the beginning of this paper. The subject was also covered in his paper on overstrain in metals.

As Mr. Gurney had remarked, the difficulties encountered by the Ordnance Department of the U. S. Army in the matter of springs were quite real, thus emphasizing the urgent need of research.

[1] Engineer of Tests, Fatigue of Metals Investigation, University of Illinois.

[2] Bureau of Standards.

[3] Ordnance Office U. S. A. Mem. A.S.M.E.

Mechanical Design of Weymouth Power Station

By E. W. NORRIS,[1] BOSTON, MASS.

THE Weymouth Power Station of The Edison Electric Illuminating Company of Boston is located on an estuary on the southern side of Boston Bay where deep water is available for docking ocean-going coal carriers and for condenser circulating water.

The geographical location involves a comparatively long haul on all fuel, and it is therefore expedient to use a high-grade bituminous coal in order to insure the largest heat value per unit of cost. This coal is fired on underfeed stokers under large boilers with economizers and the generation of power is effected by turbo-generators of comparatively large unit capacities. The high cost of fuel justifies the installation of heat-conserving equipment, such as economizers, extraction heaters, surface-type air coolers, etc. The thermal design of the plant is therefore based on these fundamental considerations.

Choice of Steam Pressure and Temperature

The most important feature of a thermal design is the choice of the steam pressure and temperature to be used in the plant. At the Weymouth Station a careful study was made of the features which affected these considerations, and these are quoted from a paper on this subject by Messrs. Moultrop and Pope before the American Institute of Electrical Engineers, June, 1923.[2]

When suitable economizers are employed the influence of steam pressure and temperature upon the efficiency of steam generation is negligible, but the influence of these factors upon the efficiency with which the heat in the steam can be utilized is in any case of definite importance.

The selection of the temperature at which the steam should be supplied to the turbine is almost entirely a problem of materials, as the beneficial effect of superheat, although largely indirect, continues in worthwhile proportion up to the maximum temperature which our engineering materials will properly withstand. In some isolated instances European engineers have employed temperatures as high as 750 deg. to 800 deg. fahr., and in America temperatures as high as 725 deg. fahr. have been chosen by some designers. In considering this matter for the Weymouth Station, it seemed best to place the nominal limit at 700 deg. fahr. It, however, appeared entirely practicable to employ this limiting temperature regardless of the pressure adopted.

Decision as to the most advantageous steam pressure could not be reached without a much more involved study, especially in view of the general lack of experience with the higher pressures and a decided variation in prophecy as to the net advantage to be derived through their use.

It was apparent that this study, to be complete, would have to cover the following considerations:

1 The improvement in economy of heat utilization made theoretically possible by increased steam pressure.

2 The extent to which available equipment could be expected to take advantage of the theoretical possibilities.

3 The probable effect of increased pressure upon reliability of operation and its flexibility in meeting anticipated load conditions with high overall economy.

4 Freedom in taking advantage of future changes in the art.

5 The economic balance of carrying charges and operating costs over a long period.

The development of one part of the study to determine the effect of the higher pressures upon improved economy is illustrated by the curves in Fig. 2. All of these curves have absolute steam pressure as their abscissae.

Curve No. 1 shows the total heat in one pound of steam at a uniform temperature of 700 deg. fahr. above 79 deg. fahr., which is the temperature corresponding to one inch absolute pressure. It will be noted that the total heat shows a decrease with increasing pressure.

Curve No. 2 shows the heat remaining in the steam after perfect adiabatic expansion from the stated initial conditions to a pressure of 1 in. absolute. The vertical distance between this curve and curve No. 1 accordingly represents the B.t.u. per pound of steam theoretically available for doing work.

Curve No. 3 is a plotting of the available heat as a percentage of the total initial heat shown by curve No. 1 and represents the efficiency of the Rankine cycle at varying pressure. Its upwardly convex curvature indicates how the rate of increase in theoretical efficiency diminishes with increasing pressure.

Curve No. 4 shows the best efficiency at present to be expected of turbo-generators in converting the available heat, as shown by curve No. 3, into useful electrical energy. In determining this curve the unit is credited with all heat recovered in the condensate by bleeding at two stages.

Curve No. 5 is the product of curves No. 3 and No. 4 and indicates, for the different pressures, the percentage of the total initial heat in the steam which would be actually converted into electrical energy or returned to the boiler in the condensate. It will be noted that this curve takes the shape of a dome with its highest point corresponding to a steam pressure of about 600 lb. absolute.

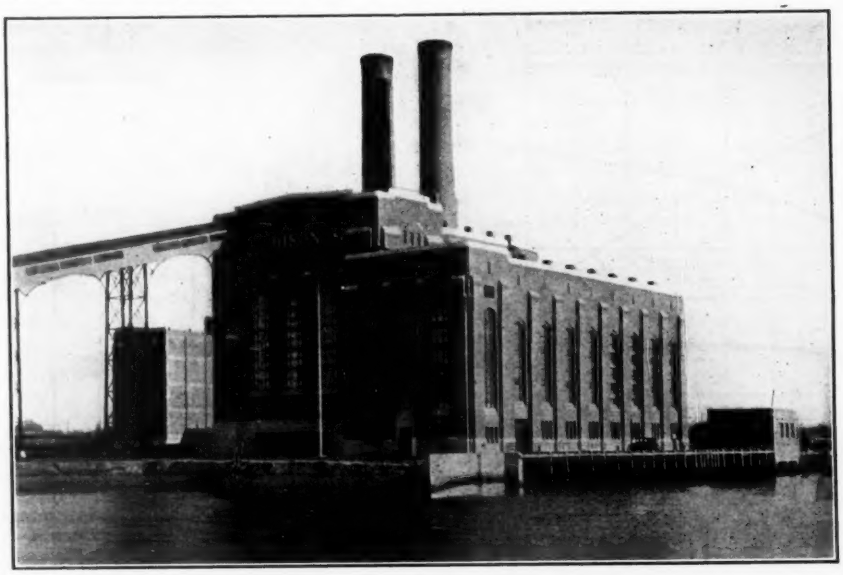


Fig. 1 Weymouth Power Station of the Edison Electric Illuminating Company of Boston

It is then seen that from thermal considerations 600 lb. would be the most efficient steam pressure. Commercial standards and capital charges modify this value, lowering the pressure to approximately 350 lb. This value was adopted for the normal steam pressure at Weymouth, in conjunction with a total temperature of 700 deg.

The loss of turbine efficiency at higher pressures can be greatly reduced by removing the steam from the turbine shell at a point near the saturation temperature, resuperheating it, and returning it to the turbine for further expansion. This reheating process greatly reduces the moisture in the lower stages of the turbine and permits a marked increase in plant efficiency at very high pressures. The elaboration of the cycle and the lack of experience with ultra high-pressure steam make it of questionable value for variable-load operation. For constant-load operation, however, there is no doubt but that the high-pressure reheating system offers desirable economies.

The Boilers

For these reasons it was decided to install a high-pressure plant that would carry a large part of the base load. After consultation with manufacturers it was found that equipment was available for use at 1000 to 1200 lb. pressure, and this value was accordingly adopted. The high-pressure plant consists of a single boiler of approximately 17,000 sq. ft. of surface with a unit economizer, both to operate at a maximum pressure of 1200 lb. per sq. in. and deliver steam to a turbo-generator of 3150 kw. (100 per cent power factor) which turbine exhausts through a resuperheater located in the boiler to the main steam header. This section of the plant will generate approximately 3000 kw. in the high-pressure turbine and approximately 12,000 kw. additional in the normal-pressure turbines.

[1] Mechanical Division, Stone & Webster, Inc. Mem. A.S.M.E.

[2] *Journal A.I.E.E.*, August, 1923, pp. 799–808.

Presented before the Boston Local Section of The American Society of Mechanical Engineers, Boston, Mass., Mar. 26, 1925.

The design of the boilers for the Weymouth Station was based on a conception differing slightly from the usual. It was assumed that the furnace formed the fundamental consideration and that the boiler and economizer would operate as two tandem units for absorbing the heat generated in the furnace. With this end in view, the stoker capacity and dimensions were first determined and the boilers and economizers designed to absorb the heat generated so as to supply the requirements of the turbo-generators. These considerations resulted in the selection of cross-drum, vertical-header boilers, 48 tubes wide and 17 tubes high, fired by 16-retort, 25-tuyere underfeed stokers. The boilers are equipped with economizers having 66 per cent of the boiler surface. This arrangement gives an exit flue gas of low temperature but avoids economizer-tube sweating.

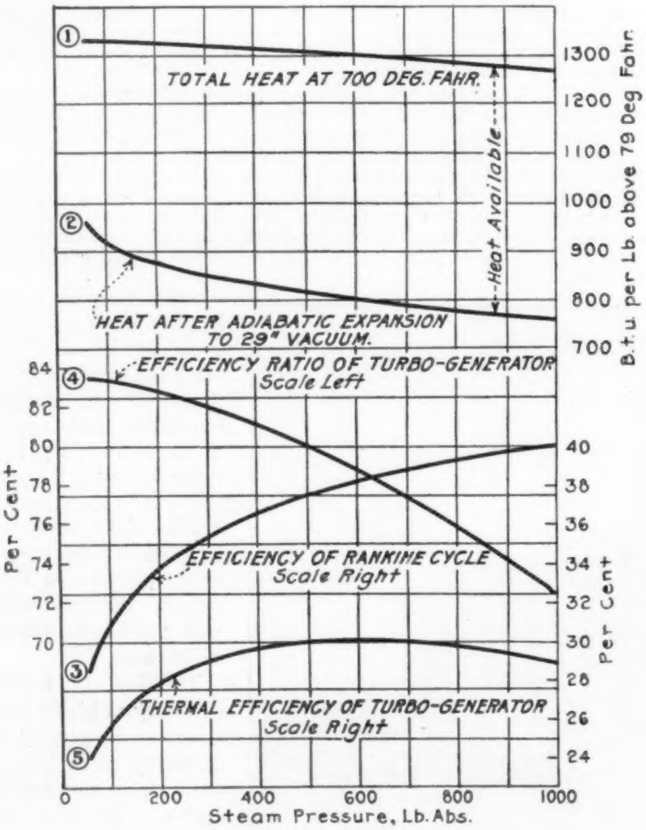


FIG. 2 STEAM-PRESSURE AND EFFICIENCY CURVES

The main turbo-generators are of the straight impulse type, having 17 stages, operating at a throttle pressure of 350 lb. at 700 deg. fahr. total temperature. The turbines are arranged for extraction to feedwater heaters at two points, namely, the 12th and 15th stages. Each turbine is of 32,000 kw. capacity and drives a 30,000-kw. 80 per cent power factor main generator, and a 2000-kw. 80 per cent power factor auxiliary generator which furnishes power for all auxiliary unit drives. The main units operate at 1800 r.p.m.

The auxiliaries are, as far as possible, grouped so as to insure convenience in operation and unrestricted access for maintenance work. The main pumping units, excluding the circulating and hotwell pumps, are located in an auxiliary bay, served by a light crane immediately below a pipe gallery which carries all headers for general service.

The Feedwater Circuit

In laying out the feedwater circuit, advantage was taken of possibilities for heat absorption from various sources. The condensate, after leaving the main-condenser hotwells, passes through a number of heat-transfer units arranged in such sequence that heat is absorbed in direct relation to the temperature rise of the condensate. Deaerators are installed in the circuit to eliminate air and dissolved gases which might cause corrosion in the boilers. The quantity of water is regulated by a make-up and draw-off system in conjunction with high-pressure evaporators so that the water circuit can be kept balanced at all times.

In laying out the details of this system, use was made of flow diagrams arranged to clearly indicate the course of the water through the various pieces of heat-transfer apparatus.

The flow from the main hotwells to the entrace of the low-pressure extraction heaters is shown in Fig. 3 and is laid out for one main unit. Leaving the hotwell pumps, the condensate passes through surface-type air coolers under the main and auxiliary generators.

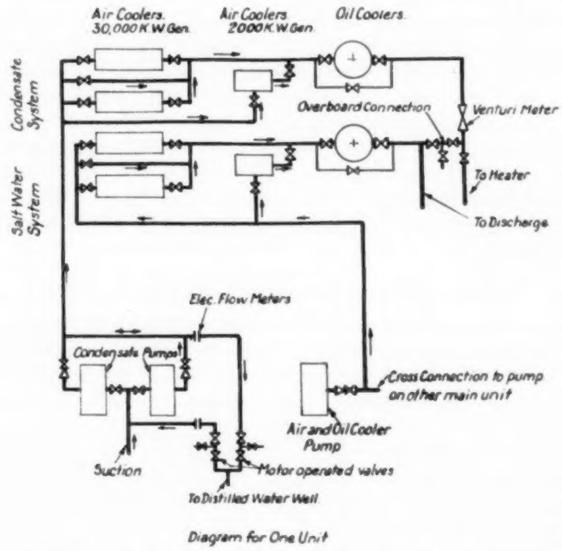


FIG. 3 FLOW DIAGRAM FOR AIR AND OIL COOLERS

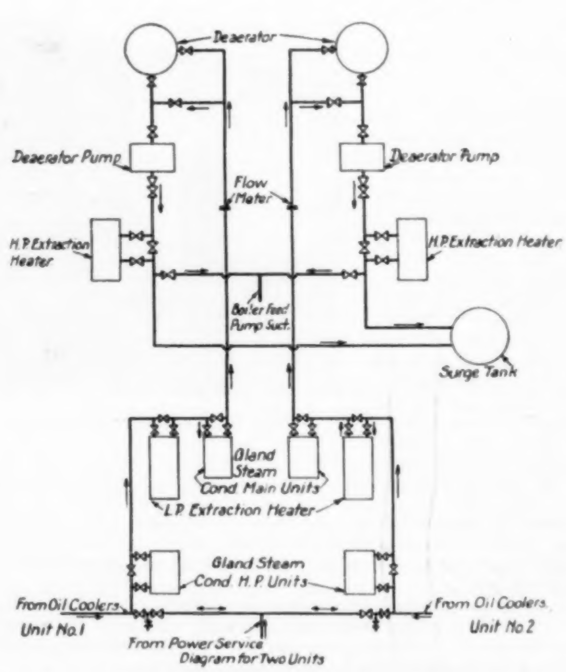


FIG. 4 FLOW DIAGRAM, OIL COOLERS TO SURGE TANK

In this way the heat losses from the generators are transferred to the condensate and returned to the boilers. From the air coolers the water passes through oil coolers, absorbing the heat generated in the main turbo-generator bearings. As the condensate quantity is greatly reduced at very light loads, auxiliary air and oil coolers are provided to take care of generator and bearing cooling at these times. These auxiliaries are supplied with salt water from the condenser circulating system, but delivered by a special pump so as to maintain the system in operation even under non-condensing conditions. A bypass is provided from the condensate discharge to the salt-water discharge so that should the condensate be contaminated with salt water through leaking condenser tubes, it

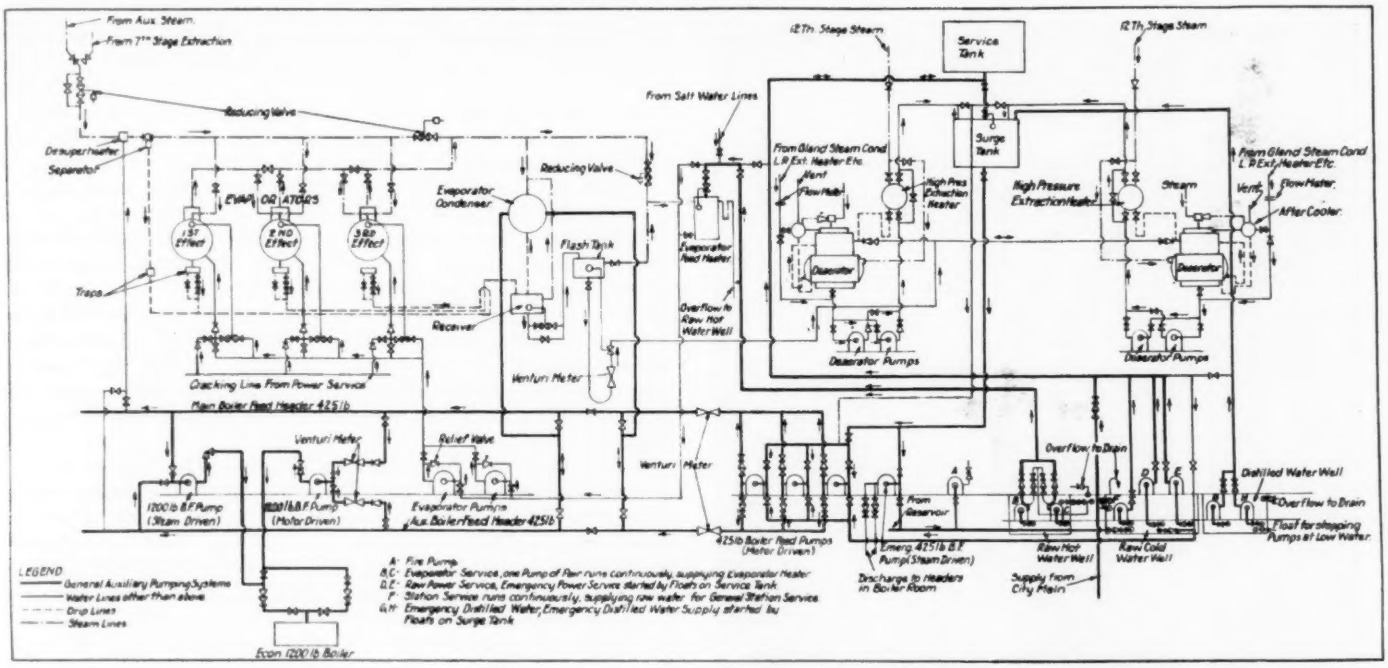


Fig. 5 Flow Diagram for Auxiliaries

may be discharged overboard and raw water pumped through the heating system to supply the necessary boiler feed.

The flow, for two main units, from the oil-cooler discharge to the surge tank is shown in Fig. 4. The water flows through the gland-steam condensers of the high-pressure turbines into the low-pressure extraction heaters, then through the normal gland-steam condensers to the deaerators. From the deaerator hotwells it is pumped through the high-pressure extraction heaters to the surge tank.

The low-pressure and high-pressure extraction heaters, as the names imply, receive steam from the 15th and 12th stages of the turbine to which they are connected, the condensed steam being returned to the boiler-feed system. The deaerator operates with a steam supply drawn from the extraction line which supplies the high-pressure extraction heater.

The gland-steam condensers are closed heaters used to condense the steam leaking from the high-pressure glands of the main and high-pressure turbines. The normal gland-steam condensers are located between the low-pressure extraction heaters and the deaerators so as to raise the temperature of the water without interfering with the operation of the extraction heaters. As there is not sufficient temperature gradient to take care of the gland-steam condenser connected to the glands on the high-pressure turbine, which operates in conjunction with one of the normal-pressure turbines, it is located between the oil cooler and the low-pressure extraction heater. In this way, when the gland-steam condenser is not in operation, it does not interfere with the heat circuit of the main unit; and when placed in operation it partially supersedes the low-pressure extraction heater, thereby condensing the gland leakage from the high-pressure turbine without unbalancing the water-heating system.

The surge tank is a hot-water storage tank used, as the name implies, to give flexibility to the system and to provide a storage of water on the suction side of the boiler-feed pumps. Contamination

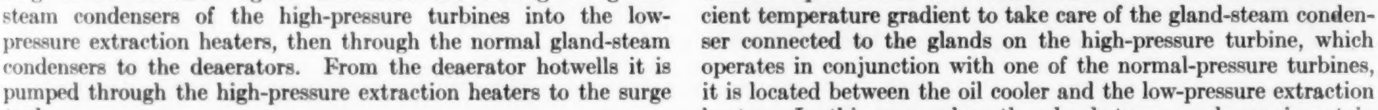


Fig. 6 Flow Control Diagram

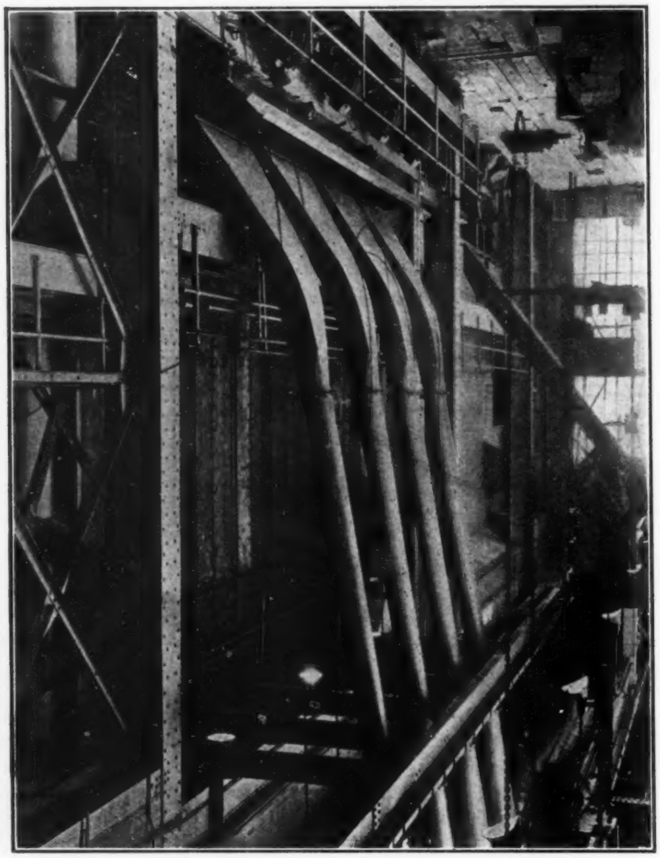


FIG. 7 BOILER-ROOM FIRING AISLE

by air is prevented by maintaining a steam blanket above the water in the surge tank at all times. The diagram shows the connection from the power service system for supplying raw water to the heater system when condensate is being discharged overboard. This connection is provided with two valves with a drip between so that leakage may be detected and prevented.

The connections for the auxiliary pumps, the evaporator system, and the boiler-feed pumps are shown in Fig. 5. The water system in general comprises a raw cold-water well from which water is pumped to an elevated service tank, which supplies general service for power and house use. The condensate return and boiler-feed system are balanced by the surge tank already mentioned and a distilled-water well. The elevated tanks are supplied by float-controlled pumps which deliver water from the wells to the tanks as required. The surge tank is normally floating on the boiler-feed system, and the distilled-water pumps are therefore used only in emergency. The service tank is kept filled by a small, constantly running centrifugal pump, which is relayed by two larger-capacity pumps for emergency service. A special arrangement is provided for balancing the make-up system.

The Flow Control System

The flow control system is shown in Fig. 6. The make-up for boiler feed is derived from a three-effect high-pressure evaporator which discharges hot water directly into the feedwater system at the deaerators. As this water may be in excess of the requirements of the system, valves are provided on the discharge and on the suction of the main-condenser hotwell pumps so that water may be drawn off or added, as required to balance the system. These draw-off and make-up connections are controlled by motor-operated valves with push-button stations located on the flow control board. They allow water to be discharged to or returned from the distilled-water well.

The flow control board comprises a series of panels on which are mounted gages for indicating the water level in the various tanks; push-button stations and switches for manually or automatically controlling the emergency pumps both for distilled and raw water, and on separate panels, a series of electrically operated flow meters and valve controls for indicating the flow of water to and from the system, and controlling the valves, as explained above. In connection with the floats which operate the distilled- and raw-water pumps, signal lights mounted on a small panel at the right of the flow control board indicate the level of the water in the service and surge tanks and form a check on the water-level gages.

The Auxiliary Equipment

In the boiler house the auxiliary equipment includes the forced- and induced-draft fans, the stoker drive, and the control of the several units. The boilers are arranged in two rows paralleling the axis of the plant, with a central aisle between used for operation, and firing aisles on the outer sides. The stoker-operating mechanism is located in the firing aisles with large concrete coal bunkers above, as the central aisle is used for operation and is shut off from the firing aisles by light partitions built across all transverse aisles.

Forced-draft fans for both rows of boilers are located on the ground floor under the firing aisle farthest from the turbine room. They supply air to the stoker wind boxes through ducts which run alternately to the near and far rows of boilers. Each boiler has an individual fan. Flexibility is obtained by the installation of a transfer duct opening through dampers into all cross-ducts.

The induced-draft fans are located immediately above the boilers and beside the economizers, each boiler having a single induced-draft fan. The central control aisle is kept clear from the operating floor to the skylights in the roof so that excellent lighting and ventilation are secured.

The control equipment is of the electrical relay type. Each boiler has an individual control board on which are mounted all the instruments necessary to indicate and record the complete operation of the boiler. These instruments are furnished with contacts which regulate the fans and stokers through relays. The boiler

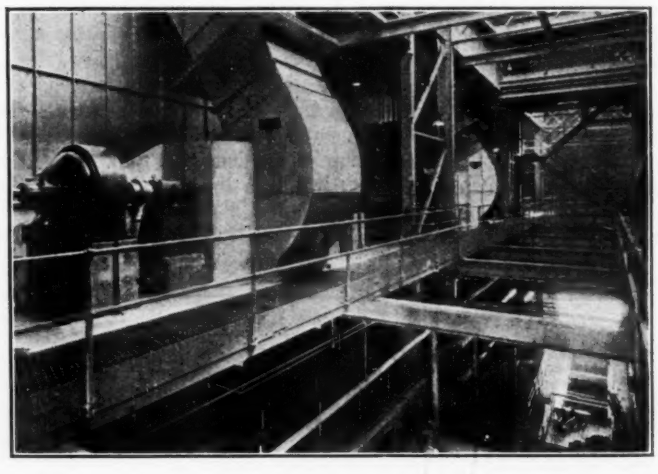


FIG. 8 INDUCED-DRAFT-FAN FLOOR

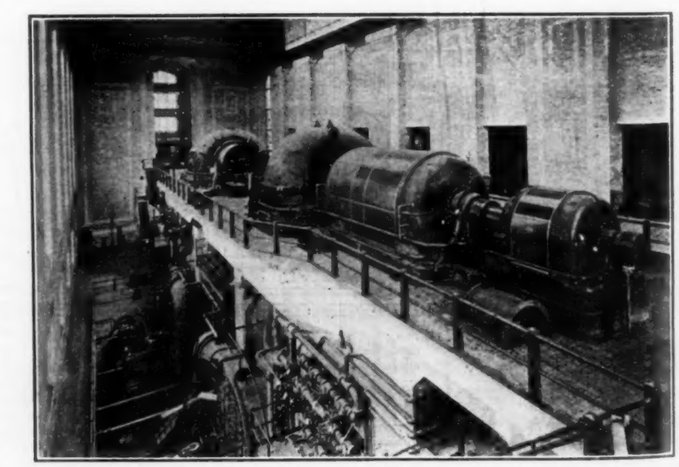


FIG. 9 MAIN-TURBINE PLATFORM

room as a whole is under the control of the master regulator which controls the boilers as a group so as to maintain constant steam pressure.

As the theoretical design for the plant was carried through, the physical design was correspondingly developed. The following illustrations will show how the actual plant corresponds to the flow diagrams and other considerations which form its foundation.

One of the firing aisles of the boiler room is represented in Fig. 7, which shows the coal chutes from the bunkers to the stokers and the stoker-drive motors which operate each stoker in four sections. The location of the boiler headers is indicated by the access doors about the center of the picture.

The induced-draft fans and the upper part of the central control aisle are shown in Fig. 8 The economizer for one of the boilers can be seen back of the induced-draft fan in the upper left-hand part of the picture, while the operating front of the next boiler can be seen at the lower right hand. The photograph gives an excellent idea of the uniformity of lighting secured.

The turbine platform with the auxiliary equipment grouped beneath it is shown in Fig. 9. The arrangement of condensers and circulating pumps is clearly indicated. The valves in the lower central part of the picture are grouped on the water boxes of the generator ventilating air coolers. The platforms between the two turbines below the main operating platforms are used for extraction-heater support. This photograph gives an idea of the actual installation of the equipment indicated in Fig. 3.

The low-pressure extraction heaters and the gland-steam condensers for two units are illustrated in Fig. 10. This equipment corresponds to that shown in Fig. 4. It should be noted that the heater nozzles have been arranged to give the minimum piping run and are equipped with manifold valves so as to avoid exterior by-passes. The gages and dial-type thermometers relating to each heater and condenser are grouped on small gage boards immediately above.

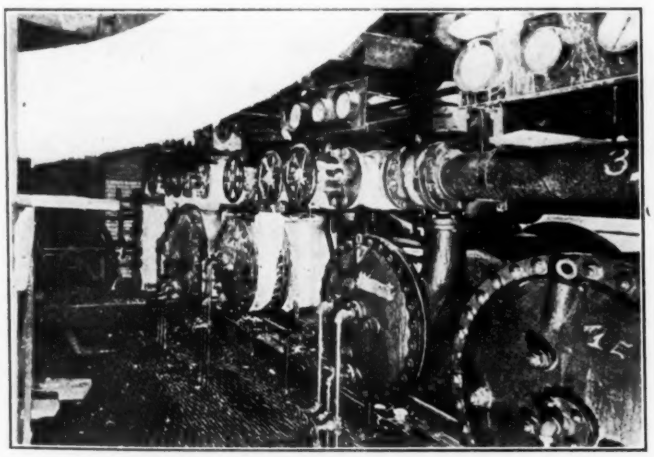


Fig. 10 Extraction Heaters

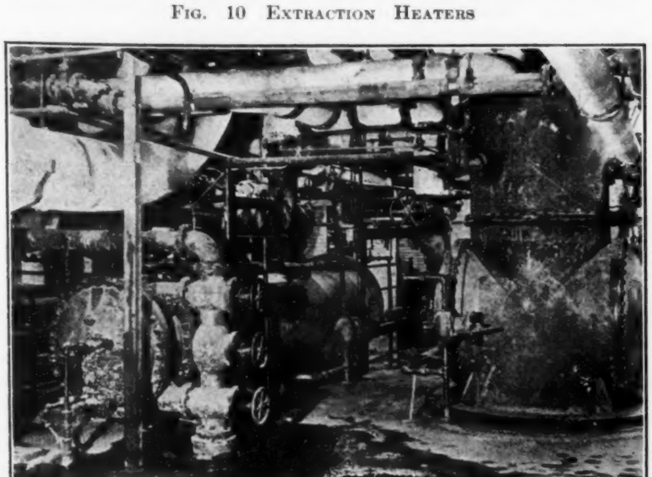


Fig. 11 Deaerating Equipment

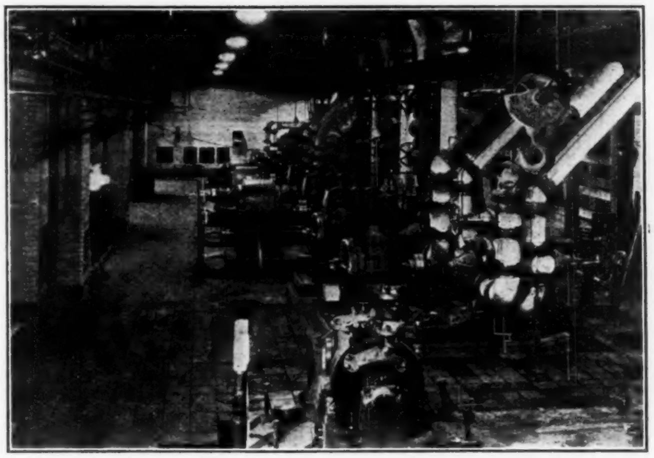


Fig. 12 Main Pumping Aisle

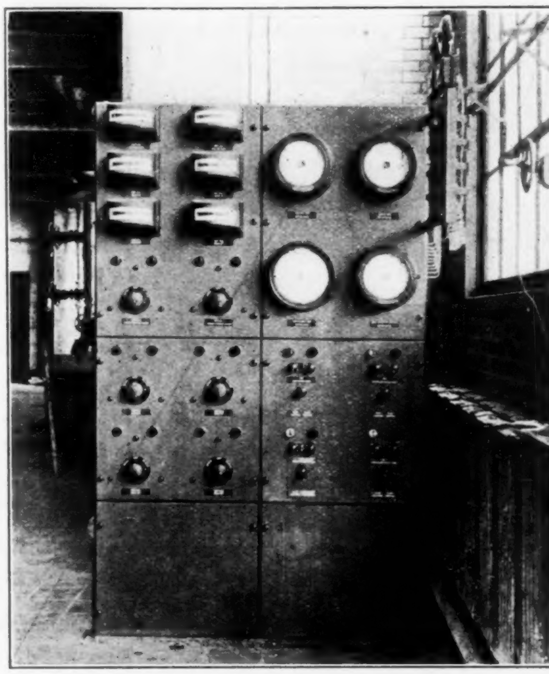


Fig. 13 Flow Control Board

In Fig. 11 the deaerators are shown at the right and the high-pressure extraction heaters at the left. This equipment is located in the auxiliary bay on a mezzanine above the pump aisle. The pipe gallery is to the left of the equipment shown. This illustration shows additional equipment indicated in Fig. 4.

A general view of the pump aisle showing the boiler-feed and deaerator pumps in the center is given in Fig. 12. Evaporator pumps are located at the far end of the aisle. The piping connections to the pumps are carried back of the crane rail to the headers in the pipe gallery which is immediately above the pump aisle. The equipment shown is that indicated in Fig. 5.

A general view of the flow control board appears on Fig. 13, it illustrates the equipment shown on Fig. 6.

The improvement in power-station engineering, which has been increasingly rapid of late years, is based very largely on a growing appreciation of the connection between theory and practice. It is to illustrate this connection as it appeared in designing the Weymouth Power Station that this discussion has been written.[1]

[1] In addition to the article in the *Journal A.I.E.E.*, already mentioned, interesting material on this subject will be found in *Power*, vol. 61, no. 15, April 14, 1925, pp. 561–567. A cross-section of the station, drawn to a large scale, and comprehensive data on the principal equipment are given on a supplementary sheet to that issue. Attention may also be called to articles in *Power Plant Engineering*, vol. 28, no. 15, Aug. 1, 1924, pp. 805–806, 2 figs., and *Electrical World*, vol. 85, no. 16, April 18, 1925, pp. 809–815.—Editor.

Deflection of a Shaft at the Critical Speed

By JOHN A. DENT,[1] LAWRENCE, KAN.

IN THE ordinary discussions of the theory of deflection of a shaft under the centrifugal force of the loads carried, it is assumed that steady running conditions exist. This theory leads to the conclusion that at the so-called "critical speed" the deflection should be infinite. In an admirable paper E. L. Thearle[2] takes up the study of the transient phenomena of bending of a shaft carrying a single concentrated load while the shaft is being brought up to speed. The conclusion there reached is that, if the shaft passes through critical speed with an angular acceleration α, the deflection at critical speed is finite and inversely proportional to α.

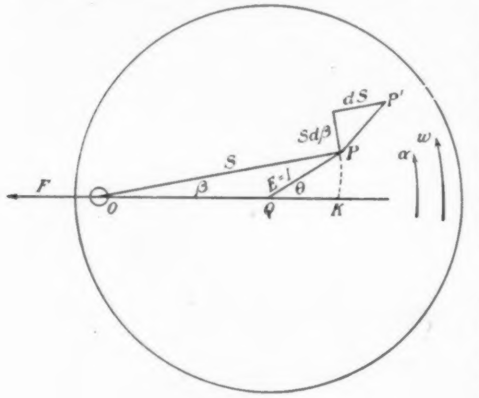


FIG. 1

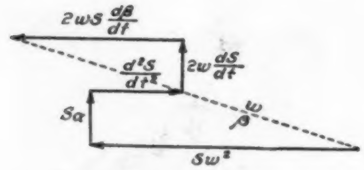


FIG. 2

In the paper referred to, certain assumptions are made which are equivalent to neglecting some components of the acceleration of the center of gravity of the disk. The present paper is an attempt to discuss the effect of the neglected components on the deflection of the shaft at critical speed, and with the further assumption that the shaft is uniformly accelerated.

In Fig. 1 let O be the center of rotation of the shaft, Q the center of the shaft passing through the disk, and P the center of gravity of the disk. Also, let

$QP = E =$ eccentricity of the disk (assumed as unity for convenience)
$OQ = Y =$ deflection of the shaft
$OP = S =$ radius of rotation of P
$\omega =$ angular velocity
$\alpha =$ angular acceleration, and
$\omega_c =$ critical velocity.

After an interval of time dt, suppose P has moved relatively to O into a new position P'. The components of this displacement are:

dS in a radial direction $\qquad\qquad S d\beta$ tangentially

The components of the velocity of P relative to a plane rotating about O with the angular velocity and acceleration of the disk are:

$$\frac{dS}{dt} \text{ radially} \qquad S\frac{d\beta}{dt} \text{ tangentially}$$

As the only external force acting on the disk is the elastic force F furnished by the shaft, the acceleration of the center of gravity P is in the direction of F. The value of this acceleration is easily shown to be

$$a = Y\omega_c^2 \quad \ldots\ldots\ldots\ldots [1]$$

where Y is the deflection of the shaft.

In the paper referred to above the acceleration of P was taken to consist of two components:

(1) A radial component $S\omega^2$ in the direction PO

(2) A tangential component $S\alpha$ perpendicular to PO.

In reality there are also

(3) A radial component $\frac{d^2S}{dt^2}$ in the direction OP

(4) The so-called "Coriolis" component perpendicular to PP'. The latter may be resolved into

(4a) $2\omega S\frac{d\beta}{dt}$ in the direction PO, and

(4b) $2\omega\frac{dS}{dt}$ perpendicular to PO.

These components are shown in Fig. 2. The resultant a is horizontal.

The vector sum of the components (1) to (4b) is the resultant acceleration $Y\omega_c^2$ in the direction QO. To define a vector such as $Y\omega_c^2$ it is sufficient to give its components parallel to any two lines in its plane. Usually, but not necessarily, these lines are chosen at right angles to each other. In the present case it is found to be more convenient to express the components in the radial and vertical directions.

The radial components are:

$$Y\omega_c^2\cos\beta = S\omega^2 - \frac{d^2S}{dt^2} + 2S\omega\frac{d\beta}{dt} \quad \ldots\ldots\ldots [2]$$

The vertical components vanish, or

$$S\alpha\cos\beta + 2\omega\frac{dS}{dt}\cos\beta + \frac{d^2S}{dt^2}\sin\beta - 2\omega S\frac{d\beta}{dt}\sin\beta - S\omega^2\sin\beta = 0 \quad [3]$$

From the geometry of Fig. 1,

$$Y = OQ = OK - QK = S\cos\beta - \sqrt{1 - S^2\sin^2\beta} \quad \ldots [4]$$

which, substituted in Equation [2], gives

$$S\omega^2 - \frac{d^2S}{dt^2} + 2S\omega\frac{d\beta}{dt} = S\omega_c^2\cos^2\beta - \omega_c^2\cos\beta\sqrt{1 - S^2\sin^2\beta} \,.\, [5]$$

From Fig. 1,

$$S\sin\beta = \sin\theta \quad \ldots\ldots\ldots\ldots [6]$$

$$\sin\beta = \frac{1}{S}\sin\theta \quad \ldots\ldots\ldots\ldots [7]$$

$$\cos\beta\frac{d\beta}{dt} = \frac{S\cos\theta\frac{d\theta}{dt} - \sin\theta\frac{dS}{dt}}{S^2} \quad \ldots\ldots\ldots [8]$$

According to Thearle's paper already referred to, at critical speed $\theta = 90$ deg. Assuming this value as a working hypothesis to be verified later, we have at critical speed:

$$\cos\beta\frac{d\beta}{dt} = -\frac{1}{S^2}\frac{dS}{dt} \quad \ldots\ldots\ldots\ldots [9]$$

Substituting this value in Equations [5] and [3],

$$S\omega^2 - \frac{d^2S}{dt^2} - \frac{2\omega}{S\cos\beta}\frac{dS}{dt} = \omega_c^2\cos\beta \times \left(S\cos\beta - \sqrt{1 - S^2\sin^2\beta}\right) \,.. [10]$$

$$S\alpha\cos\beta + 2\omega\frac{dS}{dt}\cos\beta + \frac{d^2S}{dt^2}\sin\beta + \frac{2\omega}{S}\frac{dS}{dt}\tan\beta - S\omega^2\sin\beta = 0 \,.. [11]$$

Since at critical speed S and ω are both rather large quantities, the terms $\frac{2\omega}{S\cos\beta}\frac{dS}{dt}$ and $\frac{2\omega}{S}\frac{dS}{dt}\tan\beta$ can be neglected in comparison with the terms $S\omega^2$ and $S\omega^2\sin\beta$. Equations [10] and [11] then reduce to

[1] Associate Professor of Mechanical Engineering, University of Kansas, Mem. A.S.M.E.

[2] The Rotating Disk. By E. L. Thearle. MECHANICAL ENGINEERING, Nov., 1924, p. 670.

$$\frac{d^2S}{dt^2} = S\omega^2 - S\omega_c^2 \cos^2\beta + \omega_c^2 \cos\beta(1 - S^2 \sin^2\beta)^{1/2} \quad [12]$$

$$\frac{d^2S}{dt^2} = S\omega^2 - S\alpha \cot\beta - 2\omega \frac{dS}{dt} \cot\beta \quad [13]$$

Equations [12] and [13] involve four unknown quantities: S, $\frac{dS}{dt}$, $\frac{d^2S}{dt^2}$, and β. It was assumed that at critical speed $\sin\beta = \frac{1}{S}$, but this assumption is still to be verified. Differentiating Equations [12] and [13] twice gives four more equations with two additional unknowns, $\frac{d^3S}{dt^3}$ and $\frac{d^4S}{dt^4}$. Thus a system of six equations with six unknowns is set up. Four of the unknowns can be readily eliminated, leaving two equations in S and β. (See Appendix No. 1.)

The two final equations are

$$\frac{\omega_c^2\beta^2}{S^3}\left[\frac{\beta}{2\omega}\left\{S\omega_c^2\left(1-\frac{\beta^2}{2}\right) - S\omega_c^2\left(\frac{1}{S^2}-\beta^2\right) - \frac{S\alpha}{\beta}\right\}\right] +$$
$$\left(\frac{1}{S^2}-\beta^2\right)^{3/2}\left[2\frac{\omega^3}{\beta} - \frac{6\omega\alpha\left(1-\frac{\beta^2}{2}\right)}{\beta^2}\right]\left[\frac{\beta}{2\omega}\left\{S\omega_c^2\left(1-\frac{\beta^2}{2}\right) -\right.\right.$$
$$\left.\left. S\omega_c^2\left(\frac{1}{S^2}-\beta^2\right) - \frac{S\alpha}{\beta}\right\}\right] + \left(\frac{1}{S^2}-\beta^2\right)^{3/2}\frac{4\omega^2\alpha S}{\beta} = 0 \quad [14]$$

and

$$\frac{\beta}{2\omega_c}\left[S\omega_c^2\left(1-\frac{\beta^2}{2}\right) - S\omega_c^2\left(\frac{1}{S^2}-\beta^2\right)^{1/2} - \frac{S\alpha}{\beta}\right]$$
$$\left[\left(\frac{1}{S^2}-\beta^2\right)^{1/2}\left\{\frac{3\alpha}{\beta} - \omega_c^2\left(1-\frac{\beta^2}{2}\right) - \frac{4\omega^2\left(1-\frac{\beta^2}{2}\right)}{\beta^2}\right\}\right.$$
$$\left. - \omega_c^2\beta^2\right] + \left(\frac{1}{S^2}-\beta^2\right)^{1/2}\left(S\omega_c^2 - S\alpha\frac{1-\frac{\beta^2}{2}}{\beta}\right) = 0 \quad [15]$$

The time has now come to check the assumption that at critical speed $\sin\beta = \frac{1}{S}$. Since β is a small angle we may write $\sin\beta = \beta = \frac{1}{S}$. If this value substituted for β in both Equations [14] and [15] leads to the same result for S, the assumption is shown to be sound. Putting $\beta = \frac{1}{S}$ and $\omega = \omega_c$, both Equations [14] and [15] reduce to

$$S\omega_c^2\left(1-\frac{1}{2S^2}\right) - S^2\alpha = 0 \quad [16]$$

Since S is a large quantity, $\frac{1}{2S^2}$ can be neglected and $S = \frac{\omega_c^2}{\alpha}$.

The interesting feature of this result is that it checks exactly with the value found by Thearle for the deflection at critical speed, although in his analysis the two components of acceleration $\frac{d^2S}{dt^2}$ and $2\omega\frac{dS}{dt}$ were neglected. The explanation is found in the fact that at critical speed both $\frac{dS}{dt}$ and $\frac{d^2S}{dt^2}$ vanish. (See Appendix No. 2.)

The determination of S and β for other speeds than the critical involves a very long and tedious cut-and-try process. No numerical results have been worked out. In general, it may be said that for values below critical speed the term $2\omega\frac{dS}{dt}$ will be added to the tangential acceleration $S\alpha$ and so will probably increase the effect of the latter. In other words, below critical speed Thearle's values for the deflection are probably too high. Above critical speed $\frac{dS}{dt}$ is negative. Therefore the term $2\omega\frac{dS}{dt}$ tends to neutralize the effect of the tangential acceleration $S\alpha$, and Thearle's values are probably too low. The symmetrical curves of deflection in Thearle's Fig. 7 should probably be replaced by curves which are lower at the left and higher at the right.

In the foregoing discussion it has been assumed that the acceleration α was constant. If it is assumed that the deflection will be maximum at critical speed, the value of this deflection can be found without making any assumption as to the character of the acceleration, and with much less mathematical difficulty than is involved above. (See Appendix No. 3.)

Conclusion

Thearle's analysis gives a satisfactory mathematical explanation of the fact that the shaft does not break when carried rapidly through the critical speed. His values for the deflection at critical speed appear to be correct. Just before critical speed the deflection probably builds up more rapidly than the Thearle curves would indicate, and after passing through critical speed the shaft probably returns to normal less rapidly than these curves show.

The whole field of transient phenomena of bending of shafts lies open for further mathematical and experimental investigation.

APPENDIX NO. 1

$$\frac{d^2S}{dt^2} = S\omega^2 - S\omega_c^2 \cos^2\beta + \omega_c^2 \cos\beta(1 - S^2 \sin^2\beta)^{1/2} \quad [12]$$

$$\frac{d^2S}{dt^2} = S\omega^2 - S\alpha \cot\beta - 2\omega \frac{dS}{dt} \cot\beta \quad [13]$$

Differentiating both equations with respect to t and remembering that $\frac{d\omega}{dt} = \alpha$ (assumed constant),

$$\frac{d^3S}{dt^3} = 2\omega\alpha S + \omega^2\frac{dS}{dt} - \omega_c^2\cos^2\beta\frac{dS}{dt} - \omega_c^2\sin^2\beta\cos\beta \times$$
$$\left(\frac{1}{S^2}-\sin^2\beta\right)^{-1/2}\frac{dS}{dt} \quad [18]$$

$$\frac{d^3S}{dt^3} = 2\omega\alpha S + \left(\omega^2 - \alpha\cot\beta\right)\frac{dS}{dt} - 2\alpha\cot\beta\frac{dS}{dt} - 2\omega\cot\beta\frac{d^2S}{dt^2} \quad [19]$$

Differentiating Equations [18] and [19],

$$\frac{d^4S}{dt^4} = 2\alpha^2S + 4\omega\alpha\frac{dS}{dt} + \omega^2\frac{d^2S}{dt^2} - \omega_c^2\cos^2\beta\frac{d^2S}{dt^2} - \left[\frac{\omega_c^2\sin^2\beta\cos\beta}{S^3}\times\right.$$
$$\left.\left(\frac{1}{S^2}-\sin^2\beta\right)^{-1/2}\left(\frac{dS^2}{dt}\right)\right] - \omega_c^2\sin^2\beta\cos\beta\left(\frac{1}{S^2}-\sin^2\beta\right)^{-1/2}\frac{d^2S}{dt^2} \quad [20]$$

$$\frac{d^4S}{dt^4} = 2\alpha^2S + 4\alpha\omega\frac{dS}{dt} + \left(\omega^2 - \alpha\cot\beta\right)\frac{d^2S}{dt^2} - 4\alpha\cot\beta\frac{d^2S}{dt^2} -$$
$$2\omega\cot\beta\frac{d^3S}{dt^3} \quad [21]$$

Here are six equations with six unknowns: S, $\frac{dS}{dt}$, $\frac{d^2S}{dt^2}$, $\frac{d^3S}{dt^3}$, $\frac{d^4S}{dt^4}$ and β.

Eliminating all the derivatives of S and reducing:

$$\left[\frac{\omega_c^2\sin^2\beta}{S^3}\left(\frac{1}{S^2}-\sin^2\beta\right)^{-3/2}\right]\left[\frac{\sin\beta}{2\omega}\left\{S\omega_c^2\cos\beta - S\omega_c^2\times\right.\right.$$
$$\left.\left.\left(\frac{1}{S^2}-\sin^2\beta\right)^{1/2} - \frac{S\alpha}{\sin\beta}\right\}\right]^2 + \left[\frac{2\omega^3}{\sin\beta} - \frac{6\omega\alpha\cos\beta}{\sin^2\beta}\right]\left[\frac{\sin\beta}{2\omega}\left\{S\omega_c^2\cos\beta -\right.\right.$$
$$\left.\left.S\omega_c^2\left(\frac{1}{S^2}-\sin^2\beta\right)^{1/2} - \frac{S\alpha}{\sin\beta}\right\}\right] + \frac{4\omega^2\alpha S}{\sin\beta} = 0 \quad [22]$$

$$\left[\frac{\sin\beta}{2\omega}\left\{S\omega_c^2\cos\beta - S\omega_c^2\left(\frac{1}{S^2}-\sin^2\beta\right)^{1/2} - \frac{S\alpha}{\sin\beta}\right\}\right]\left[\frac{3\alpha}{\cos\beta} -\right.$$
$$\left.\omega_c^2\cos\beta - \omega_c^2\sin^2\beta\left(\frac{1}{S^2}-\sin^2\beta\right)^{1/2} - \frac{4\omega^2\cos\beta}{\sin^2\beta}\right] +$$
$$\frac{2\omega}{\sin\beta}\left(S\omega^2\ S\alpha\cot\beta\right) = 0 \quad [23]$$

Equations [22] and [23] contain two unknowns, S and β. The form of these equations makes a general algebraic solution impossible, and even the determination of simultaneous values of S and β for assumed values of ω, ω_c, and α is a very laborious and tedious process. At the critical speed, however, some results can be relatively easily found. If $\omega = \omega_c$ and β is a small angle so that we may write

$$\sin\beta = \beta$$

$$\cos\beta = 1 - \frac{\beta^2}{2}$$

Equations [22] and [23] become, after multiplying through by $\left(\frac{1}{S^2}-\beta^2\right)^{1/2}$

and $\left(\frac{1}{S^2} - \beta^2\right)^{1/2}$, respectively,

$$\frac{\omega_c^2\beta^2}{S^3}\left[\frac{\beta}{2\omega}\left\{S\omega_c^2\left(1-\frac{\beta^2}{2}\right) - S\omega_c^2\left(\frac{1}{S^2}-\beta^2\right)^{1/2} - \frac{S\alpha}{\beta}\right\}\right]^2 + \left(\frac{1}{S^2}-\beta^2\right)^{3/2}\left[\frac{2\omega_c^3}{\beta} - \frac{6\omega\alpha}{\beta^2}\left(1-\frac{\beta^2}{2}\right)\right]\left[\frac{\beta}{2\omega}\left\{S\omega_c^2\left(1-\frac{\beta^2}{2}\right) - S\omega_c^2\left(\frac{1}{S^2}-\beta^2\right)^{1/2} - \frac{S\alpha}{\beta}\right\}\right] + \left(\frac{1}{S^2}-\beta^2\right)^{3/2}\frac{4\omega^2\alpha S}{\beta} = 0 \qquad [24]$$

and

$$\left[\frac{\beta}{2\omega}\left\{S\omega_c^2\left(1-\frac{\beta^2}{2}\right) - S\omega_c^2\left(\frac{1}{S^2}-\beta^2\right)^{1/2} - \frac{S\alpha}{\beta}\right\}\right]\left[\left(\frac{1}{S^2}-\beta^2\right)^{1/2} \times \left\{\frac{3\alpha}{\beta} - \omega_c^2\left(1-\frac{\beta^2}{2}\right) - \frac{4\omega_c^2(1-\beta^2/2)}{\beta^2}\right\} - \omega_c^2\beta^2\right] + \left(\frac{1}{S^2}-\beta^2\right)^{1/2}\left(S\omega_c^2 - S\alpha\frac{1-\beta^2/2}{\beta}\right) = 0 \qquad [25]$$

To test the assumption that at critical speed $\beta = \frac{1}{S}$, substitute this value in Equations [24] and [25]. Both equations are found to reduce to

$$S = \frac{\omega_c^2}{\alpha}\left(1 - \frac{1}{2S^2}\right) \qquad [26]$$

$\frac{1}{2S^2}$ is negligible compared with 1. Therefore

$$S = \frac{\omega_c^2}{\alpha} \qquad [27]$$

APPENDIX NO. 2

Eliminating $\frac{d^4S}{dt^4}$, $\frac{d^3S}{dt^3}$ and $\frac{d^2S}{dt^2}$ from Equations [11], [12], [18], [19], [20], and [21], it can be shown that

$$\frac{dS}{dt} = \frac{\sin\beta}{2\omega}\left[S\omega_c^2\cos^2\beta - S\omega_c^2\left(\frac{1}{S^2} - \sin^2\beta\right)^{1/2} - \frac{S\alpha}{\sin\beta}\right] \qquad [28]$$

Substituting the critical values, $\omega = \omega_c$, $\sin\beta = \frac{1}{S}$, $\cos\beta = 1 - \frac{1}{2S^2}$, Equation [28] becomes

$$\frac{dS}{dt} = \frac{\beta}{2\omega_c}\left[S\omega_c^2\left(1 - \frac{1}{2S^2}\right) - S^2\alpha\right] \qquad [29]$$

Since $\frac{1}{2S^2}$ is very small and since $S = \frac{\omega_c^2}{\alpha}$,

$$\frac{dS}{dt} = \frac{\beta}{2\omega_c}\left[S\omega_c^2 - S^2\alpha\right] = \frac{\beta S}{2\omega_c}\left[\omega_c^2 - S\alpha\right] = 0 \qquad [30]$$

At critical speed, therefore, the radial velocity $\frac{dS}{dt} = 0$ and the Coriolis component vanishes. From Equation [12]

$$\frac{d^2S}{dt^2} = S\omega^2 - S\alpha\cot\beta - 2\omega\frac{dS}{dt}\cot\beta = S\omega^2 - S\alpha\frac{1-\beta^2/2}{\beta} - 2\omega\frac{dS}{dt}\cot\beta \qquad [31]$$

At critical speed $\beta = \frac{1}{S}$, $\frac{\beta^2}{2}$ is negligible, $\frac{dS}{dt} = 0$, and $S = \frac{\omega_c^2}{\alpha}$. Substituting these values,

$$\frac{d^2S}{dt^2} = S\omega_c^2 - S^2\alpha = S(\omega_c^2 - S\alpha) = 0 \qquad [32]$$

Therefore the radial acceleration $\frac{d^2S}{dt^2}$ vanishes at critical speed.

APPENDIX NO. 3

Since β is a small angle at and near critical speed, we may write

$$\sin\beta = \beta$$

$$\cos\beta = 1 - \frac{\beta^2}{2}$$

Then Equations [2] and [5] become

$$S\alpha\left(1 - \frac{\beta^2}{2}\right) + 2\omega\frac{dS}{dt}\left(1 - \frac{\beta^2}{2}\right) - 2S\omega\beta\frac{d\beta}{dt} - S\omega^2\beta + \beta\frac{d^2S}{dt^2} = 0 \qquad [33]$$

$$S\omega^2 - \frac{d^2S}{dt^2} + 2S\omega\frac{d\beta}{dt} = \omega_c^2\left[S(1-\beta^2) - \left(1 - \frac{\beta^2}{2}\right) \times \sqrt{1 - S^2\beta^2}\right] \qquad [34]$$

As β is small and S is large a first approximation is obtained by neglecting β^2 in comparison with 1 and $\sqrt{1 - S^2\beta^2}$ in comparison with S.

Then

$$S\omega^2 - \frac{d^2S}{dt^2} + 2S\omega\frac{d\beta}{dt} = \omega_c^2 S \qquad [35]$$

$$S\alpha + 2\omega\frac{dS}{dt} - 2S\omega\beta\frac{d\beta}{dt} - S\omega^2\beta + \beta\frac{d^2S}{dt^2} = 0 \qquad [36]$$

From [35]

$$\frac{d\beta}{dt} = \frac{S\omega_c^2 - S\omega^2 + \frac{d^2S}{dt^2}}{2S\omega} \qquad [37]$$

Substituting this value in [36] and reducing

$$S\alpha + 2\omega\frac{dS}{dt} - \beta S\omega_c^2 = 0 \qquad [38]$$

$$\beta = \frac{\alpha}{\omega_c^2} + \frac{2}{\omega_c^2}\,\frac{\omega dS/dt}{S} \qquad [39]$$

Differentiating,

$$\frac{d\beta}{dt} = \frac{1}{\omega_c^2}\frac{d\alpha}{dt} + \frac{2}{\omega_c^2}\left[\frac{\omega}{S}\frac{d^2S}{dt^2} + \frac{\alpha}{S}\frac{dS}{dt} - \frac{\omega}{S^2}\left(\frac{dS}{dt}\right)^2\right] \qquad [40]$$

If it be assumed that the deflection is maximum at critical speed, then when $\omega = \omega_c$

$$\frac{dS}{dt} = 0$$

Substituting these values in Equations [37] and [40],

$$\frac{d\beta}{dt} = \frac{1}{2S\omega_c}\frac{d^2S}{dt^2} = \frac{1}{\omega_c^2}\frac{d\alpha}{dt} + \frac{2}{S\omega_c}\frac{d^2S}{dt^2} \qquad [41]$$

$$\therefore \quad \frac{d^2S}{dt^2} = -\frac{2S}{3\omega_c}\frac{d\alpha}{dt} \quad \text{approx.} \qquad [42]$$

and

$$\frac{d\beta}{dt} = -\frac{1}{3\omega_c^2}\frac{d\alpha}{dt} \quad \text{approx.} \qquad [43]$$

Substituting these approximate values in Equation [34] and reducing, a second approximation is reached, giving

$$-\frac{S\alpha^2}{\omega_c^4} = \left(1 - \frac{\alpha^2}{2\omega_c^4}\right)\sqrt{1 - S^2\frac{\alpha^2}{\omega_c^4}} \qquad [44]$$

Solving

$$S^2 = \frac{\omega_c^4}{\alpha^2} - 1 \quad \text{approx.} \qquad [45]$$

Since ω_c^4 is usually large compared with α^2

$$S = \frac{\omega_c^2}{\alpha} - \frac{\alpha}{2\omega_c^2} \quad \text{approx.} \qquad [46]$$

This differs from the value previously found by the term $-\frac{\alpha}{2\omega_c^2}$. This term is usually very small compared with $\frac{\omega_c^2}{\alpha}$.

It should be noted that this result does not involve any assumption as to the constancy of the acceleration α.

Example. Let the running speed of the shaft be 400 radians per sec. and its critical speed be 200 radians per sec. While gaining speed the value of ω will be some function of the time t.

Assume that the speed may be expressed by a function of the form $\omega = 400(1 - e^{-mt})$. This expression indicates that the shaft starts with high acceleration and that the acceleration rapidly decreases as the running speed is approached.

If the value of m be taken as 0.7 the shaft will reach its critical speed in 0.99 sec.—a very high rate of acceleration.

Then at critical speed

$$\omega = 400(1 - e^{-0.7t}) = 200 \quad \therefore \quad e^{-0.7t} = 0.5$$

$$\alpha = \frac{d\omega}{dt} = 0.7 \times 400\, e^{-0.7t} = 140$$

$$\frac{d\alpha}{dt} = -0.49 \times 400\, e^{-0.7t} = -98$$

Then from Equation [46]

$$S = \frac{200^2}{140} - \frac{140}{2 \times 200^2} = 285$$

From Equation [42]

$$\frac{d^2S}{dt^2} = -\frac{2 \times 285}{3 \times 200} \times 98 = -93$$

$$S\omega_c^2 = 285 \times 200^2 = 11{,}400{,}000$$

showing that at critical speed the radial acceleration $\frac{d^2S}{dt^2}$ is quite negligible compared with the centripetal acceleration $S\omega_c^2$.

Since it is not likely that a shaft will ever be brought up to speed more rapidly than in this example, it follows that at critical speed the radial acceleration can be neglected if the maximum deflection is attained at this speed.

The assumption of maximum deflection at critical speed implies that the radial velocity $\frac{dS}{dt}$ vanishes, and that therefore the Coriolis component $2\omega\frac{dS}{dt}$ also disappears.

The Parallel Operation of Hydro and Steam Plants

Some Operating and Economic Features That Should Receive Careful Attention

By F. A. Allner,[1] Baltimore, Md.

Due to the low generating costs of modern steam plants the margin of saving offered by a hydro development is frequently quite small. The utmost utilization of the combined hydro-steam source of power has become of greater economic importance in recent years. The economic service of hydro power is usually expressed as replacement value of the equivalent amount of steam power, comprising the steam investment cost for the hydro demand service and the steam generating costs for the hydro energy delivered to the load system.

A hydro development that is not equipped with seasonal storage can render the greatest demand service to a given system load if, on the days of greatest steam demand on the system coinciding with minimum flow, the hydro plant supplies the peaks and the steam plant the base of the system load. The greatest energy contribution from the hydro plant is obtained by letting it carry the base portion of the system load during high flow and having the peak portion produced by steam.

Methods are explained in this paper by which these economic principles are put into practice. Load signaling by means of small changes in frequency above or below normal has been found very useful for a more perfect distribution of load during certain hours when it is otherwise difficult to secure the desired division.

Hydro-steam parallel operation may produce a favorable operating combination as these two sources of power can supplement each other to mutual advantage. Hydro units are generally more reliable than steam units. They can be brought up to full load from standstill in a small fraction of the time that is required for safe operation of steam units. Troubles at hydro stations usually give sufficient forewarning to permit calling in steam assistance to make up for threatened capacity reductions. Inherent limitations which prevent too short a governor traversing time on hydro units are explained, with oscillograms of "load on" and "load off" tests recording speed, gate travel, and pressure in wheel pit. Steam units give better frequency regulation for sudden changes in load. Thus a load system that is served by mixed hydro-steam supply can obtain during a great part of the time certain service advantages of both sources of power without being subjected to the imperfections of either. The hydro-steam combination may at times offer an opportunity of better localizing the disturbing effect of city distribution trouble and of securing a more positive selective relay action than if power were supplied only from one source.

Successful parallel operation of hydro and steam plants, so far as it refers merely to maintaining synchronism under normal conditions and through the usual abnormal occurrences, presents no particular difficulties. However, in order to obtain the most favorable economic results and the best possible service conditions under definite limitations, a number of seemingly subordinate features should be given careful attention.

Few undeveloped water-power sites within transmission distance from the important load centers of the east are so endowed by nature that their power can be delivered to the city at a cost materially lower than that of local modern steam plants. The efficiency of prime-mover and fuel-burning equipment has increased in recent years. Fuel prices have generally been lowered. However, economic comparison with hydro power cannot be set down once for all in definite figures, as the cost of steam power is subject to changes that cannot be forecast for a lengthy period. After the hydro plant is completed, however, the total cost for power remains fairly constant, although the effective service of the hydro plant will enhance in value, particularly when operated in conjunction with steam plants on a growing load system. To coördinate parallel operation will therefore become of increasing importance until the theoretical optimum is reached.

The maximum in energy output will be reached when the hydro plant can operate continuously with wide-open gates throughout the excess-flow period. The maximum of peak service can be secured from the hydro plant when the system load has grown to a point where the entire hydro-generating capacity can be utilized on peak generation under minimum-flow conditions, saving then the equivalent amount of effective steam capacity in the carrying of system loads.

Feeding the two sources of power into the same load system makes for a rather favorable operating combination, if the contractual relations under which the two plants operate do not impose any rigid dividing lines as to demand or energy. It is found quite practical to shape the contract relations in such a way that the fullest coöperation is secured, even without joint ownership.

Low-Flow Operation for Maximum Hydro Peak Service

To derive the greatest saving in investment for steam capacity from parallel operation it is necessary to determine conditions

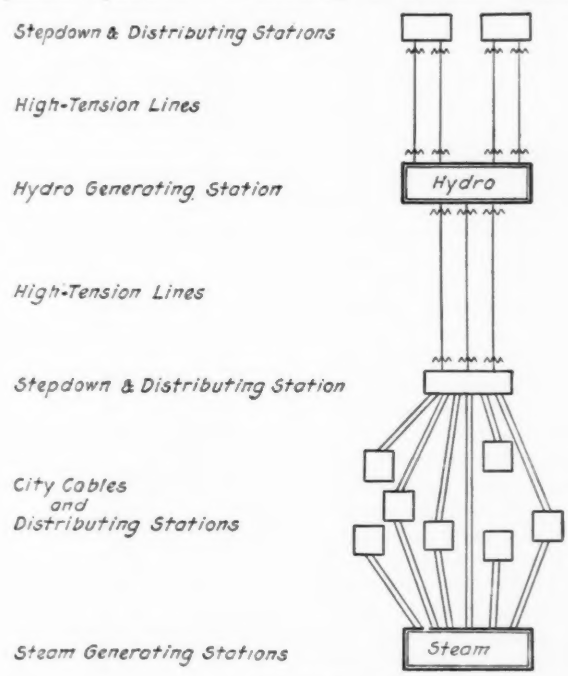


Fig. 1 System Diagram

requiring the greatest steam output; these may not occur on the day of maximum peak for the year. They will depend on the seasonal variations of system load, on variations in flow, on available storage facilities, on seasonal limitations of a given steam plant due to amount of circulating water, etc. As a rule the greatest saving in capacity investment is rendered by the hydro plant on days of maximum annual peak, which should then be assumed to coincide with the day of lowest winter flow on record. The effective steam capacity that can be thus saved may be quite considerable, even on a run-of-river development where the minimum winter flow is only a small percentage of the full-draft flow, given accurate information on flow and a reliable estimate of the system-load requirements.

Fig. 1 shows diagrammatically a typical physical layout of a high- and low-tension network supplied from the two sources of power. Fig. 2 shows typical high-flow conditions and Fig. 3 low-flow conditions, both being plotted from actual load data on representative days. The loads are drawn in as one-hour integrations, taken on the hour.

For the same year in which the actual load of Fig. 3 occurred, the maximum anticipated winter peak was placed at about 175,000 kw. The lowest hydro energy available during the winter season from the connected run-of-river development was approximately 300,000 kw-hr., neglecting any use of storage except for the purpose of equalizing the power-house draft during the 24 hours. The

[1] Gen. Supt., Pennsylvania Water and Power Company. Mem. A.S.M.E.

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peak service obtainable therefrom would have been somewhat in excess of 40,000 kw. The same hydro plant, when fully developed to a delivered capacity of about 80,000 kw., can deliver approximately 2,000,000 kw-hr. at 100 per cent utilization factor; that is, the minimum hydro energy on the critical peak day coinciding with lowest winter flow, is only 15 per cent of the energy ultimately available at high-flow stage. This 15 per cent of energy, however, can be made to render a peak service, measured by one-hour integration, of over 40,000 kw., i.e., nearly 50 per cent of the high-flow peak service of the fully developed hydro plant. As the system load continues to grow at approximately the same load factor, there is of course a tendency for this minimum-flow peak service to increase gradually until it approaches in value the full high-flow peak service.

The one-hour integration is not a complete expression of the effective peak service rendered to the load system, as on the critical peak day the steam plant is assumed to be operated at its maximum sustained output while the hydro plant carries the load variations of the entire system. The maximum indicated load in excess of the one-hour integration is frequently over 10 per cent above the hourly integration. For the system from which the load data in Figs. 2 and 3 were taken, this excess load was frequently found to be between 15,000 and 20,000 kw. This would raise the effective peak service rendered to the load system by the hydro plant to nearly 75 per cent of the fully developed high-flow peak service.

On the critical low-flow peak day the output of the hydro plant, expressed as kilowatt-hours delivered per cubic foot per hour of available river flow, will be slightly reduced by reason of the extremely low-load-factor generation. This slight loss of output, however, is of no moment on the critical low-flow day when maximum peak service is of paramount importance. Even on the average low-flow day this loss of output will usually be compensated for by the greater economy at the steam plant, due to its operation under an exceedingly high load factor. Thus it will be found advantageous to operate most run-of-river hydro plants during the low-flow period in a manner that will give highest load-factor generation to the steam plant, to which will be assigned the base load. In the example of Fig. 3 this method of dividing the generation has been carried out even to the extent of ceasing hydro output entirely during the off-peak hours and operating one or two units as synchronous condensers.

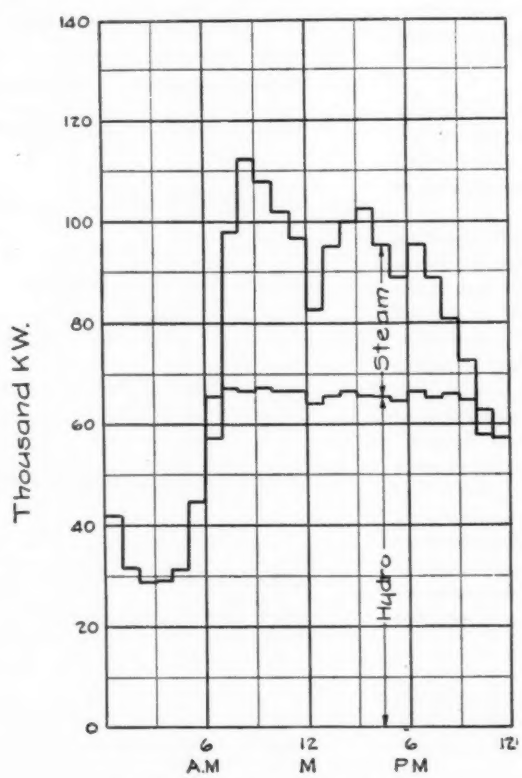

Fig. 2 Typical High-Flow Day
(Tuesday, Feb. 15, 1921.)

	Kw.	Kw-hr.	Load factor, per cent
Hydro delivered	67,200	1,368,200	84.8
Steam	45,500	442,800	40.5
Total load	112,700	1,811,000	67.4

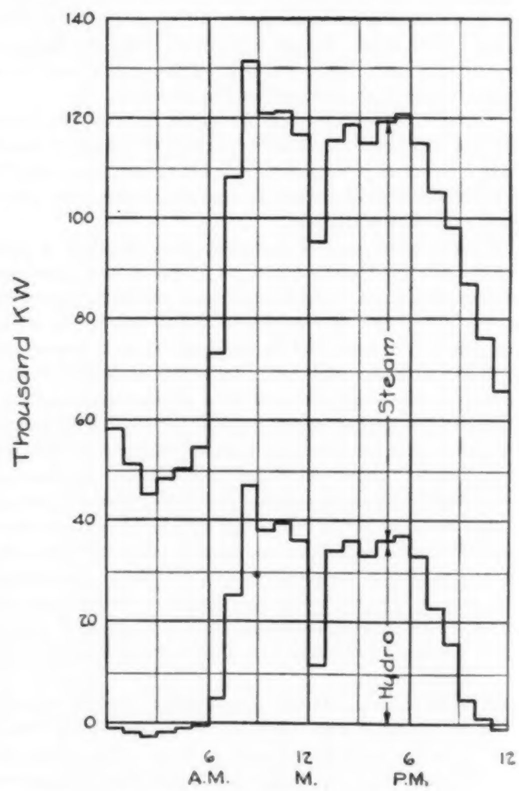

Fig. 3 Typical Low-Flow Day
(Tuesday, Oct. 24, 1922.)

	Kw.	Kw-hr.	Load factor, per cent
Hydro delivered	47,700	454,000	39.6
Steam	84,200	1,763,200	87.3
Total load	131,900	2,217,200	70.0

In the case of the typical low-flow day of Fig. 3, the reasons for this method of off-peak condenser operation were: First, the desire to let the steam plant operate even during the off-peak hours at best efficiency, and to have the hydro units come in automatically as spares on unexpected load swings or on failure of a steam unit; second, to assist the steam units in the carrying of wattless current which otherwise might require the paralleling of another steam unit, with a consequent lowering of steam efficiency; third, to protect the service of the hydro company's other customers as to capacity and also as to better selective-relay action in the event of high-tension-line disturbances; fourth, to relieve the steam plant of the voltage-regulating task for the other high-tension-line services, which can be better taken care of by field adjustments on the hydro generator operating as a synchronous condenser.

High-Flow Operation for Maximum Hydro Energy Output

During the period of excess flow the hydro plant should be operated for maximum energy output, and carry the base portion of the system load. In the load and generating curves of Fig. 2 the maximum hydro-output idea has not been carried out to the point of theoretical perfection. For conditions of less than 100 per cent utilization factor at the hydro plant, there will be at least two periods when an increase of hydro output seems theoretically possible: first, the period around the time when the ascending line of the a.m. load intersects with the hydro capacity line; second, the time of possible intersection of the hydro capacity and load lines during the noon-hour recess, i.e., between noon and 1:00 p.m.; the third period occurs when the descending line of the p.m. system load intersects with the hydro capacity line. During certain days or seasons of the year there may also be additional occasions of intersections in the afternoon and evening, before the load remains definitely lower than the hydro capacity.

During the first period the fires must be brought up rapidly and several steam units must be paralleled in quick succession to carry the rapidly increasing industrial and railway load. This point of intersection varies only within a few minutes from one weekday to another. By a careful following up of the load rise the time when additional units should be made ready for paralleling before the load demands it can be anticipated. The steam operator prefers naturally to have his boilers and prime movers on the line rather a little early than too late; he also wants to avoid just floating the unit in without load and having it motored partly during momentary downward swings of the load, unless the units are specially equipped to stand motoring.

In the example shown in Fig. 4—a large time-scale reproduction of loads between 11:30 a.m. and 1:30 p.m., including the load re-

cess during the noon hour—the minimum load during the valley of the load curve was high enough not only to load up all hydro units but also give the desired minimum load of 5000 kw. to the steam units on the line. The steam operator in this case disconnected only one of the four units from the bus, as indicated by the dotted line, letting it idle in the meantime on its own governor with reduced auxiliaries.

It is interesting to compare the high-flow load division around the noon hour shown in Fig. 4 with a corresponding large-scale illustration drawn from a typical low-flow day (Fig. 5). In the latter case the steam plant is carrying an almost flat load except for minor momentary upward and downward swings which the hydro units are unable to take over on account of the longer traversing time of their governors. All gradual and sustained variations in system load, however, are carried by the hydro plant. As hydro units can be started and stopped rather quickly and do not require

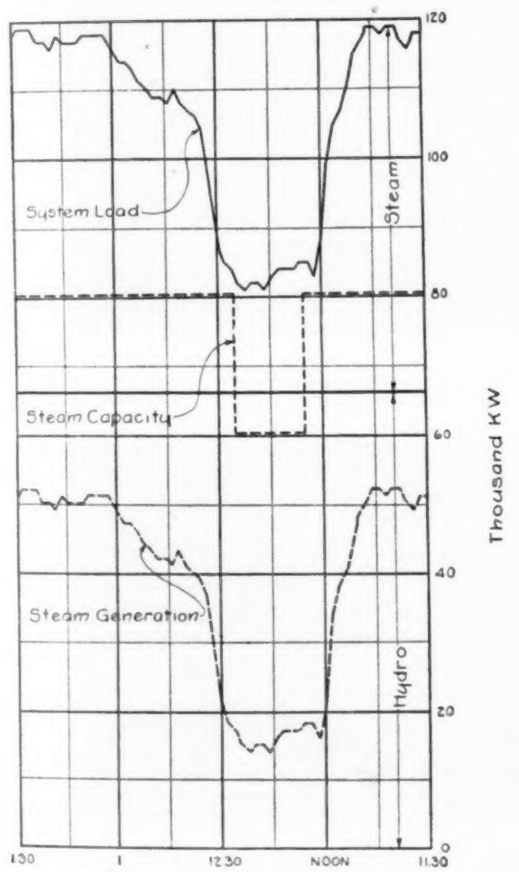

Fig. 4 Typical Load Division during Noon Recess in High Flow
(Wednesday, Mar. 14, 1923.)

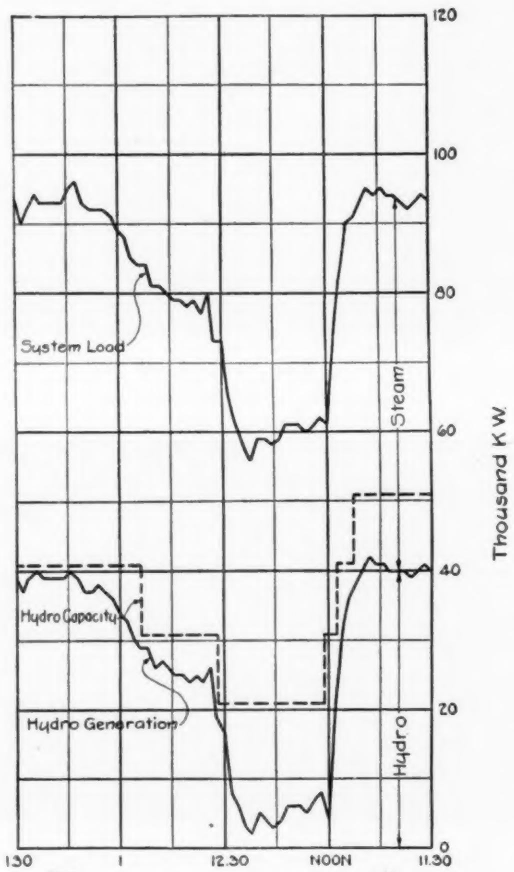

Fig. 5 Typical Load Division during Noon Recess in Low Flow
(Tuesday, Oct. 24, 1922.)

large power-consuming auxiliaries during the transition, the hydro plant can follow up the noon-hour and other sharp variations of the system load much more promptly with the cutting out and restarting of units than the steam plant, and at a very much lower sacrifice in efficiency.

During the third period the drop in load is generally not as rapid as the rise of the a.m. load or the changes at noon. The time of intersection, however, is variable, and the operators have to be as watchful as on the other occasions to avoid unnecessary loss of hydro output.

During the peak season this p.m. drop of the load is somewhat steeper and may at times call for a speedier reduction in boiler percentage rating than is consistent with good maintenance of brickwork and settings. Such a limitation should be properly recognized, even though it may serve to cause a slight loss of hydro output.

The load division during these various load-intersecting periods, was greatly improved after a load-signaling scheme by frequency indication had been adopted.[1] The hydro governors were set to maintain approximately one-half of one per cent above normal frequency whenever the plant was not fully loaded, that is, when the hydro governors maintained the frequency. When the hydro plant was wide open, that is, when the steam governors were maintaining the frequency, the latter were so adjusted that they maintained a frequency of half of one per cent below normal. Thus a glance at the frequency indicator advised the steam operator whether or not the hydro plant was fully loaded. If the frequency was above normal, he adjusted the load on the steam units by switchboard control of the synchronizing springs down to the minimum load desired on the steam unit. The hydro operator followed a converse system.

Service Features of Combined Hydro-Steam Operation

Some of the advantages of a combined hydro-steam plant from the operator's point of view warrant further discussion.

Sudden capacity reductions without warning are rather rare at the hydro plant. Average service records of large-capacity hydro units showed that the "service demand availability factor" was in excess of 99.5 per cent. This is a measure of the availability of the unit during the time in which it is needed. The fact that on individual units it sometimes is 100 per cent indicates that all maintenance and inspection work can be done while the unit is not in demand. An average taken from the service record of large steam units showed a "service demand availability factor" of approximately 92.5 per cent. This bears out the long-accepted idea that it is not nearly as necessary to provide spare capacity at a hydro plant as it is at the steam plant.

It is not safe to bring up a large steam unit from standstill to full load in much less than one hour. A hydro unit of nearly equal capacity can be brought up to full load from standstill in less than one minute. By proper training of personnel such speedy starting up becomes a regular and safe procedure. Thus the feeling of security as to carrying of the system load is greatest when one or more hydro units are held in reserve, which is the usual low-flow condition. Even if the hydro plant is shut down during the offpeak hours in the low-flow period and no units are floated in as synchronous condensers, the hydro source of power can be made quickly available to meet emergencies at the steam plant. Conversely, the steam plant cannot meet a sudden capacity shortage when the hydro plant is wide open, unless some reserve capacity is actually floating on the line.

Nearly all the capacity reductions at the hydro plant approach gradually, permitting additional steam capacity to be called in early enough to protect the system load. They occur nearly always during the high-flow period when the steam demand is well below the maximum capacity available. These conditions are caused by reduction of head due to rain or ice floods, obstructions of intake

[1] This plan was first suggested in 1919 by F. E. Ricketts, Supt. of Operation, Consolidated Gas, Electric Light and Power Company, Baltimore, Md.

screens by trash, and ice trouble in its various forms. Most capacity reductions at the steam plant come rather suddenly and without warning. Among the sudden capacity reductions are breakdowns of the main turbines and generators, and occasionally breakdowns of minor parts of the equipment such as high-pressure piping and boiler-room and turbine auxiliaries. If these accidents happen during the hours of wide-open turbine gates in the high-flow period, emergency assistance from the hydro plant will not be available; but, during the entire low-flow period and occasionally also during the off-peak hours in high flow when for some reason or other a small part of the system load is supplied by steam, the hydro plant will generally be able to make up such capacity shortage by speedy paralleling of hydro units, before its disturbing effect is felt by loss in frequency.

the pilot valve, large-sized connection pipes to and from the pilot valve, etc. By the shortening of the traversing time on the steam unit an additional advantage is frequently gained, due to the fact that the overspeed tripping device may not come into play in the event of large loss of load. This shortens the time of restoration of service following an electrical disturbance. Short traversing time on steam governors is particularly desired in parallel operation because the latter have a tendency to speed up higher for a given percentage of sudden load reduction than the steam units, which, however, being electrically coupled, are pulled along in synchronism. A quicker response of the steam governors will then cut down the percentage load change on the hydro plant as a whole, and will decrease the total amount of overspeed.[1]

On the hydro units there are practical as well as inherent limita-

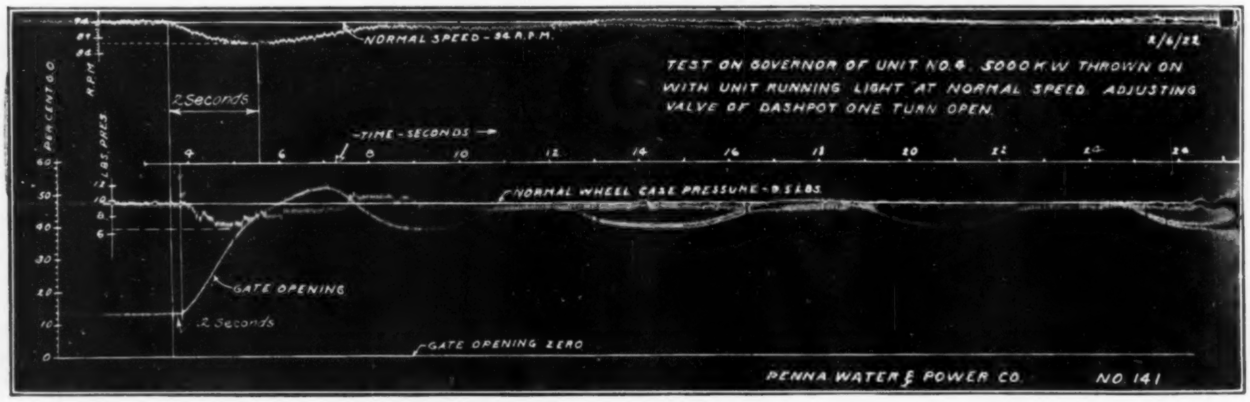

Fig. 6 Diagram of Speed, Gate Opening, and Wheel-Pit Pressure of 10,000-Kw. Hydro Unit, 53 Ft. Head, Following Sudden Throwing On of 50 Per Cent Load

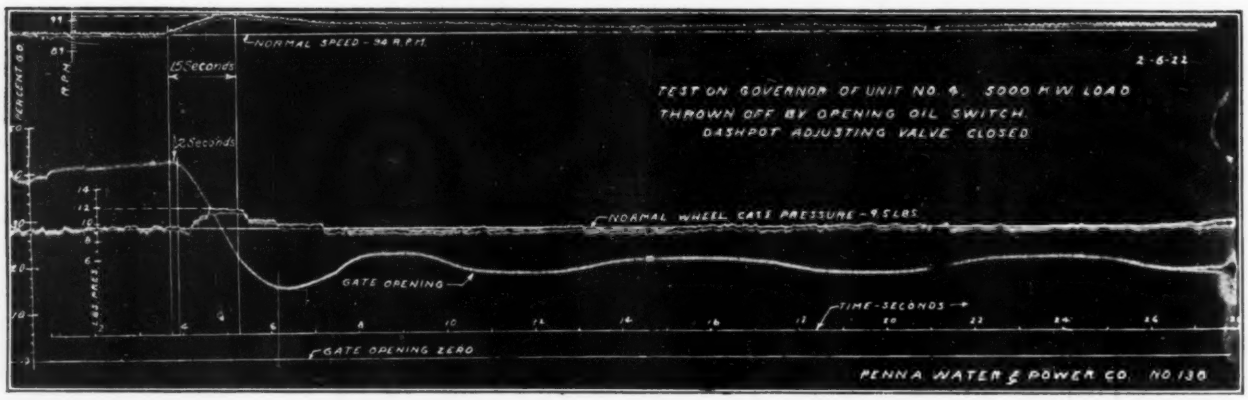

Fig. 7 Diagram of Speed, Gate Opening, and Wheel-Pit Pressure of 10,000-Kw. Hydro Unit, 53 Ft. Head, Following Sudden Throwing Off of 50 Per Cent Load

The speed regulation at times of sudden fluctuations of system loads is somewhat improved by the parallel operation with steam units as compared with all hydro generation. This is essentially due to two causes: First, because the kinetic energy per kilowatt of rated capacity that can be absorbed by the rotating masses for a given increase in speed is much greater for the high-speed steam units than for the low- and medium-speed hydro units, even though the latter have generally some extra weight added for that purpose. The second cause lies in the inherently greater total traversing time of the hydro governor.

The traversing time varies somewhat for the same unit, depending on the amount of load change, the steadiness of the governor and relay mechanism immediately before the load change, etc. "Traversing time," refers to the time between the sudden throwing off of 100 per cent load and the moment of reaching maximum speed. At the modern steam turbine this total traversing time can be held down to approximately one second. On certain types of units the standard design and adjustment may give a considerably longer time, around two seconds. It is frequently possible to reduce this traversing time by expedients such as higher setting of the relief valve in the oil-pressure supply, increase of port openings around

tions on too rapid a governor-relay action, even if it were possible to make the hydro governor and its mechanism proper as free from friction as the steam governors, and also possible to hold down the lag between the pilot valve and the relay piston. The latter would be very difficult to accomplish because of the greater multiplicity of working parts, the greater length of piping, the greater inertia of the relay fluids, etc.

Sudden reduction of water flow causes a pressure rise in the wheel casing, penstocks, etc. the water hammer of which may not only stress these parts to the breaking point but may also, on sudden gate movements, nullify the effect of reduced passages. That is, the discharge through the smaller openings at the increased pressure may impart greater power to the wheel than the discharge

[1] To the author's knowledge, A. L. Penniman, Superintendent, Steam Stations, Consolidated Gas, Electric Light & Power Company, Baltimore, was the first steam-operating engineer who emphasized the great desirability, from the service point of view, of shortening the traversing time of the standard steam-turbine governors. To Mr. Penniman should also be given credit for having originated and actually carried out on standard governors of certain types of steam turbines most of the above-mentioned expediencies, which rendered them more suitable for perfect hydro-steam parallel operation.

through the previous larger openings at normal head. This condition is illustrated in the oscillograms of Figs. 6 and 7, where wheel-pit pressure, speed, and gate travel are recorded from a 50 per cent "load on" and "load off" test made on a 10,000-kw. hydro unit operating at 53 ft. head. The effect of the rise or drop in water pressure is not completely recorded there, because in addition to the pressure changes above the wheel there are corresponding changes taking place in the draft-tube effect below the wheel, both of which work in the same direction, i.e., they impede the power-regulating effort of the governor.

The traversing time at 50 per cent load change was found to be between 1.5 and 2.0 sec. For the 100 per cent "load off" test the shortest traversing time measured on the same unit was 2.6 sec. The higher the head at the hydro plant, the greater the traversing time necessary for handling the same amount of water on a unit. Minimum values for traversing time are frequently 6 sec. and greater in plants operating at heads of 200 ft. or higher.

Effect of Hydro-Steam Parallel Operation on the Average Cost of Steam Generation

There is little doubt, that the average steam cost per kw-hr. would be higher in the case of mixed supply when the hydro plant is an all-year-round steady source of power, supplying the base portion of the system load. Inversely, the kw-hr. cost of the steam portion of the mixed supply will assuredly be lower than for all-steam supply in case a hydro plant that is equipped with seasonal storage furnishes the peak portion of the system load through the greater part of the year.

In the case of run-of-river developments, hydro-steam parallel operation, if conducted under a coöperative plan, will show a lower average steam cost during low flow but a higher cost per kw-hr. of steam generation during high flow. The overall result will probably be controlled by the weighted preponderance of the higher or lower load-factor generation compared with the load factor of the total system load. A steam station operating in parallel with a run-of-river hydro plant is subject to considerable variation of operating conditions in the course of the year and, at times, also to sudden changes from one day to the next. In order to obtain the best possible results under these conditions from the combined sources of power, the personnel at the steam plant has a more painstaking task to perform than in those stations where the entire system load is supplied by steam or where the hydro plant furnishes a fixed portion of the load throughout the year.

Seasonal Diversity in the Time of Maintenance Work for Hydro and Steam Plants

If the seasonal high-flow period occurs with regularity for several consecutive months each year, the mixed supply will offer to the steam plant an opportunity of carrying on major repair work and overhauling in an efficient manner. This lowers the unit cost of maintenance per kw-hr. and also makes possible certain savings of investment in spare parts necessary under all-steam supply to compensate for the capacity withdrawals during the period of overhauling.

A further advantage in maintenance can be realized when the hydro and steam plants are close enough to permit the utilization of the same maintenance force and of joint shop facilities for the routine overhauling work. This is favored by the natural diversity in the time of repair work for the two plants. The run-of-river hydro plant will always endeavor to have all its equipment in continuous service during the high-flow season. The high-flow period, on the other hand, means generally an easing off of capacity demand on the steam plant, and possibility of repair work on the steam plant. The low-flow season, on the other hand, is the best time in which to carry out all the major repair work at the hydro plant.

Discussion

A. G. Christie[1] wrote that steam stations connected to hydroelectric systems were becoming of increasing importance for the economical use of water power as the system load grew. It was difficult, he said, in stations of this character to make an estimate of the probable use factor for the steam equipment on which to base the economic selection of plant machinery. Engineers would be interested in figures on the use factors of the steam stations in Baltimore and of the hydro plant at Holtwood.

[1] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

C. W. Place[1] thought that the importance of peak service by hydro plants to lighter-weight systems covering a larger area would be even greater than described by the author. The paper, he said, led to the inquiry as to whether the proper way to think of hydroelectric power was not in terms of peak and off-peak power, rather than as primary and secondary, or firm and dump, power. The true value to a system, he said, was due to the improvement in load factor on the steam equipment and consequent decrease in cost of the total steam production, rather than in reducing the coal consumed in such steam production at perhaps inconvenient times.

Charles B. Hawley[2] said that in order to take advantage of operating throughout the entire year in such a way that the base load was carried by steam and the peak load by hydro plants, and to avoid duplication of hydro installation by reserve steam capacity, seasonal storage was required for the water power. For greatest economy a river should be developed as a whole, with seasonal storage located, as far as practicable, at the headwater plants, and preferably at medium- or high-head installations. He also showed by means of a lantern slide the effective service that hydro plants could render steam plants by carrying their peak loads.

W. W. Eberhardt[3] wrote that the operating conditions of the Pennsylvania Water Power Company's system and that of the Alabama Power Company were very similar, and he outlined how the Alabama Company divided its load in wet and dry seasons. The method of controlling load division between hydro and steam plants by frequency indication, as described by the author, would not be tolerated in southern systems with large cotton-mill loads because of the resultant variation in the speed of the looms. He described methods of control of power flow used by the Alabama Power Co.

W. M. White[4] said that he could not agree that steam power should be used in preference to hydro power when the difference was merely a fraction in favor of the steam, because he thought that engineers had a responsibility in developing nature's resources with a minimum of human labor. When it was considered that a 70,000-kw. hydro unit saved the labor of more than 2000 men merely to mine, prepare, transport, handle, and fire the coal necessary to produce a like amount of power by steam, it could be seen that the question of cost was not the only one affecting the choice between steam and water power.

John F. Vaughan[5] said that the paper pointed out again the difference between the fixed charges and operating costs of steam and hydro plants, and the true value of local power in a market supplied with power brought in over long distances from more favorable sources.

Wm. W. Tefft[6] wrote of his experiences with hydraulic governors, proper gate opening, and aeration of draft tubes.

In answer to Professor Christie the author wrote as follows:

"For the three years, 1921 to 1923, the average use factor at the Holtwood plant was about 55 per cent, at the Westport steam station, about 25 per cent. It must be considered, however, that the hydro plant operated, so to speak, without spare prime-mover capacity, while the steam plant had considerable spare prime-mover capacity, some of which was not operated at all. Preference was given at the steam plant to the more efficient prime-mover capacity and the use factor on this part of the equipment was comparatively high, comparing favorably for individual units with base-load conditions in the average steam station. The load factor of the total steam generation at the Westport plant, obtained as a weighted daily average, was approximately 70 per cent during the three-year period."

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[2] Consulting Engineer, Washington, D. C.
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[4] Manager, Chief Engineer, Hydraulic Dept., Allis-Chalmers Mfg. Co., West Allis, Wis. Mem. A.S.M.E.
[5] Vaughan Engineers, Boston, Mass. Mem. A.S.M.E.
[6] Vice-President and Chief Engineer, Consumers Power Co., Jackson, Mich.

A Comparison of Modified Ball-Indentation Hardness Numbers

By E. STANLEY AULT,* HOUSTON, TEX.

ALTHOUGH it is generally recognized that the static indentation test does not measure any one distinct quality of material, it is nevertheless a test that is convenient and universally established for determining the quality of hardness and the numerous other qualities that accompany it. Furthermore, the Brinell test is most widely known, and any efforts to standardize or reduce to one basis the Brinell hardness number will increase its value until a more satisfactory test is secured and established, as well as serve as a guide in the selection of that test. The United States has not been as keen as England and the continental countries in the investigation of attempted measurements of hardness, and the author presents here a brief summary of the search for a single number to express indentation hardness independently of the load and the diameter of the ball, and a comparison of the various methods when applied to accepted data.

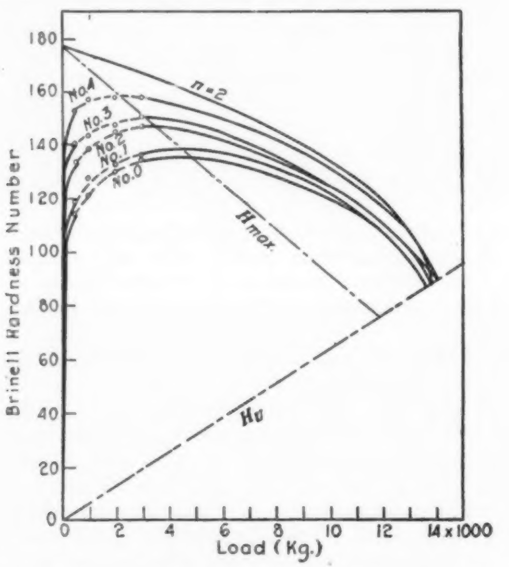

Fig. 1 Relation of Brinell Hardness Number to Load
(Baker and Russell's cold-worked steels.)

When Brinell[1] established his test, he fixed the ball diameter and load, and as long as these were adhered to the results of different tests were comparable. It is not always desirable or possible to use the standard conditions, for softer materials require a smaller ball and a smaller load than do iron and steel. It was then found that the hardness number obtained varied with the diameter of the ball and the magnitude of the load, and it was soon realized that the variation in value was not due entirely to the difference in cold working of the material under test.

Modifications Proposed by Various Investigators

Benedicks of Upsala[2] found from tests that if the Brinell number is multiplied by the fifth root of the radius of the ball, a constant number is obtained for varying ball diameters. With a load of 3000 kg., the Brinell hardness number,

$$\text{B.h.n.} = \frac{P}{A} \times \sqrt[5]{\frac{D_1}{10}}$$

where P = load employed in kg., A = spherical area of indentation in sq. mm., and D_1 = diameter of ball used in mm.

Le Chatelier[3] modified this to care for different loads, to

$$\text{B.h.n.} = \frac{P_1}{A_1} \times \sqrt[5]{\frac{D_1}{10}} \times \frac{20{,}000}{17{,}000 + P_1},$$

where P_1, A_1, and D_1 apply to the conditions of test. Both of these formulas are purely empirical and have been disproved by Batson[4] and H. Moore.[5] As the correction is given to apply to all materials, it can be no better than approximate.

In 1908, E. Meyer[6] showed from tests on 28 different metals and alloys that the mean pressure per unit of projected area ($4P/\pi d^2$) is a constant for a given angle of impression regardless of the size of the ball. He further showed that $P = ad^n$, where a and n are constants depending upon the initial state of the material and the diameter of the ball, P is the load in kg., and d the diameter of the indentation in mm.

The same year Kurth[7] confirmed these findings for nickel and copper and further suggested that n depends upon the condition of the metal as regards strain. The factor n is then a measure of the capability of the material to stand further hardening by cold work, and a is a constant for each material for a given ball diameter. The value of n is greatest when the material is in the annealed condition.

Meyer's hardness number is

$$P_m = \frac{4P}{\pi d^2} = \frac{4ad^{n-2}}{\pi}$$

and, for materials where $n = 2$, the hardness number is independent of the load, angle of impression, and diameter of the ball. When the diameter of the impression, d, is 1 mm., $a = P$ and is the load necessary to make an indentation whose diameter is 1 mm. Values for n and a can be obtained readily from two impressions of the same ball under different loads. If the logarithm of the load is

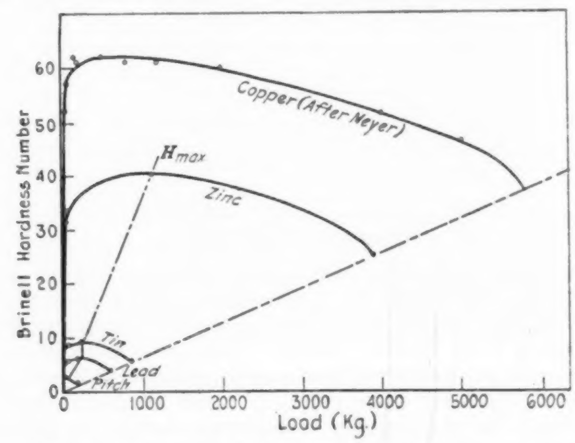

Fig. 2 Relation of Brinell Hardness Number to Load
(Non-ferrous metals and pitch.)

plotted against the logarithm of the diameter, a straight line of slope n results;

$$n = \frac{\log P_1 - \log P}{\log d_1 - \log d}$$

where d and d_1 are respectively the diameters of the indentation at loads P and P_1.

Values for n range from 2 to 2.5. Representative values given by Meyer show the variation with annealing:

Copper, hard-worked	2.01
Copper, annealed	2.39
Brass, hard-rolled	2.06
Brass, annealed	2.53

It is to be noted that n approaches 2 with cold work. O'Neill[8] found $n = 2$ for plasticine. Norbury[9] found that for lead n decreased with the time of loading, finally approaching 2.

H. Moore[5] in 1909 pointed out that if a standard angle of impression of 30 deg. be taken, the diameter of the impression would

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be equal to the radius of the ball and the number would become—

$$P_m = \frac{16\,PD^{n-2}}{\pi(2d)^n}$$

This formula was checked by Batson[4] and appears to give a constant number for any load or size of ball. As it is based upon the projected area, as is Meyer's, it must be divided by 1.0718 to correspond to the Brinell number.

Several investigators have observed that there is a certain load that gives a maximum Brinell number. This is clearly shown in Figs. 1 and 2, reproduced from O'Neill,[8] where a curve similar to a stress-strain diagram with a maximum point is obtained. True comparison of different materials is not obtained with an arbitrary load, for that particular load may pass through the point of greatest Brinell hardness in some materials and not reach it in others.

In order to express the Brinell hardness of a material by one figure, Waizenegger[10] and O'Neill[8] worked independently for a maximum Brinell number that could be computed readily from D, a, and n, but independently of P.

O'Neill first transformed the spherical area

$$A = \frac{\pi D}{2}\left(D - \sqrt{D^2 - d^2}\right)$$

by substituting for d the value $(P/a)^{1/n}$ obtained from the value $P = ad^n$. Then, dividing the expression obtained into the load, P, he obtained a general expression for the indentation number H:

$$H = \frac{2}{\pi D} \times a^{2/n} \times P^{(1-2/n)}\left[D + \sqrt{D - (P/a)^{2/n}}\right]$$

Putting the first derivative equal to zero, he obtained

$$H_{\max.} = \frac{2a}{\pi} \times D^{n-2} \times \frac{n}{n-1} \times \left[\frac{n(n-2)}{(n-1)^2}\right]^{\frac{n-2}{2}}$$

Waizenegger put his value for H in the form of

$$H = \frac{P}{\frac{\pi}{2}D^2\left(1 - \cos\frac{\phi}{2}\right)}$$

where $\phi/2 = \sin^{-1}(d/D)$. This expression is easily proved identical with O'Neill's, and consequently the values of $H_{\max.}$ are identical, although Waizenegger obtained this in the form of

$$H_{\max.} = \frac{2aD^{n-2}}{\pi} \times \frac{\left[\frac{\sqrt{n(n-2)}}{(n-1)}\right]^n}{\frac{(n-2)}{(n-1)}}$$

O'Neill expressed his equation in the form of

$$H = yP^x + yP^x \times z/D$$

where $x = 1 - \dfrac{2}{n}$, $y = \dfrac{2a^{2/n}}{\pi}$

and $z = \sqrt{D^2 - (P/a)^{2/n}}$

z becomes zero when $P = aD^n$ and the hardness number at this point, independent of the load, O'Neill calls the "ultimate Brinell hardness," or

$$H_u = yP^x = \frac{2a}{\pi} \times D^{n-2}$$

This is the hardness number when the diameter of the impression is equal to the diameter of the ball. O'Neill found that H_u seemed to remain constant with cold work and depended upon the initial condition of the material.

In 1916 W. N. Thomas[11] presented data from which he obtained, for loads above 1000 kg., a straight-line relation $P = HA - K$, where K is a constant, H the Brinell hardness number, A the spherical area. On repeated annealing and impressing on the same spot, the slope of the load-area line and its intercept diminished, and, at the point where the load just "floated," the intercept became zero. This gave $P = HA$, eliminating K, the factor of work hardening. The average value of K he found to be 300, hence,

$$H = (P + 300) \div A$$

If $3000/A_1$ be adopted as the hardness number,

$$A = \frac{P + 300}{3000 + 300}A_1$$

$$H = \frac{3000}{A_1} = \frac{3000}{\dfrac{3300}{P + 300}A} \qquad \therefore H = \frac{3000}{3300}\left(\frac{P + 300}{A}\right)$$

where H = brinell hardness number
A = spherical area, in sq. mm.
A_1 = modified spherical area to give constant Brinell hardness number
P = load applied, in kg.

Below 1000 kg. the points apparently lie upon the line $P = 106.5\,A$ passing through the origin. He found from his log load-log diameter graph $P = ad^n$, averaging $P = 50d^{2.39}$ nearly.

Harris[12] in 1922 used the repeated annealing and loading method in his so-called "strainless indentation" tests of brasses. He observed the same diminishing of K and called the final relation of P/A at the "floating state" the "absolute hardness number." He considered that the factor K, due to cold work, was eliminated by the repeated annealing, leaving finally the primitive hardness. This method of annealing the Brinell impression hardly simplifies matters as much as it seems. The amount of strain in the metal diminishes as the distance from the impression increases and the annealing at uniform temperature would promote rapid crystal growth, thus altering the mechanical condition of the metal.

Harris also stated from his data that $H = P^x + C$, and called C the hardness constant of the material, as $H = C$ at zero load. Thomson said, Greenwood concurring, "Brinell hardness equals a constant, which is a function of the elastic limit of the metal, plus a term which is a function of the stress applied, indicating the extent to which the metal was hardened by cold work." The expression $H = P^x + C$ and the definition are based upon the assumption that the Brinell-load curve does not pass through the origin when extended.

Oddly enough, Greenwood (1918)[13] in his tests upon Al-Cu alloys

TABLE 1 COMPARISON OF BALL-INDENTATION HARDNESS NUMBERS

Observer	Diam. of ball, mm.	Material	Load, kg.	Diam of indentation, mm.	Computed value of n	Computed value of a	Brinell hardness number	Moore's modified hardness number	$H_{\max.}$ of O'Neill and Waizenegger	H_u of O'Neill	Norbury's log $a + n$
O'Neill	10	Steel A: 0.25 C, 0.28 Mn	500	2.30	2.29	74	119	145	142	91	4.16
			1000	3.10			129	150			
			2000	4.20			138	151			
			3000	5.04			140	148			
			3500	5.39			141	150			
		Steel W: 14.0 W, 0.65 C 0.25 Mn, Cr, Va	500	1.53	2.29	185	270	383	354	231	4.56
			1000	2.09			288	376			
			2000	2.80			318	383			
			3000	3.37			327	374			
			3500	3.62			329	371			
		Steel 4: 0.9 C, Heat Tr.	500	1.35	2.29	262	348	507	507	328	4.71
			1000	1.79			394	530			
			2000	2.41			432	534			
			3000	2.90			444	525			
		Steel 3: 0.9 C, Heat Tr.	500	1.20	2.29	342	441	666	663	428	4.83
			1000	1.60			495	682			
			2000	2.17			534	672			
			3000	2.58			564	685			
		Steel 2N	30		2.19	112	112		117	71	4.23
		Steel S90	30		2.30	264	264		260	168	4.72
		Steel A	30		2.25	150	149		152	96	4.42
		Steel W	30		2.31	323	323		316	206	4.82
		Plastacine			1.99	0.015			0.019	0.01	0.066
		Pitch			1.96	1.83			2.3	1.17	2.22
		Lead			2.26	3.18			5.8	3.68	5.76
		Tin			2.19	5.53			8.8	5.39	2.93
		Zinc			2.21	24.0			40.1	24.8	3.59
Meyer	10	Copper	30	0.857	2.09	45	52	61	62	35	3.74
			49	1.041			57	69			
			160	1.801			62	68			
			200	2.042			61	66			
			500	3.174			62	65			
			800	4.006			61	65			
			1200	4.841			61	65			
			2000	6.184			60	66			
			4000	8.635			51	65			
			5000	9.530			46	65			

TABLE 1 (Continued)

Harris	10	Brass, 70:30	500	3.10	2.35	38	65	78	81	54	3.92
			1000	4.10			73	82			
			1500	4.80			78	83			
			2000	5.40			79	82			
			2500	6.02			80	82			
			3000	6.50			80	83			
Greenwood	10	Chill-cast Cu-Al 8.7 Al	500	2.78	2.10	76	81	88	107	61	3.98
			1000	3.62			94	100			
			2000	4.74			106	114			
			3000	5.75			105	113			
			4000	6.45			108	119			
		Same, 10. Al	500	2.38	2.53	55	111	166	174	118	4.27
			1000	3.15			125	164			
			2000	4.14			142	163			
			3000	4.85			152	164			
			4000	5.65			145	149			
		Same, 14.3 Al	1000	2.03	2.47	172	305	468	467	322	3.71
			2000	2.70			341	463			
			3000	3.20			364	456			
			4000	3.64			372	443			
Batson	4.76	Mild steel	350	1.92	2.25	81	116	129	154	76	4.15
			500	2.27			116	125			
	10		2000	4.50	2.20	73	119	129	151	74	4.06
			3000	5.41			120	127			
	4.76	Boiler plate	400	1.97	2.25	87	125	137	176	83	4.19
			500	2.17			127	138			
	10		2000	4.42	2.34	62	123	136	141	68	4.13
			3000	5.25			128	136			
	4.76	Rail steel	300	1.46	2.14	133	176	192	207	104	4.26
			400	1.67			177	192			
	10		2000	3.71	2.22	109	178	198	238	115	4.26
			3000	4.45			183	198			
	4.76	High-C steel	350	1.41	2.25	161	220	256	322	152	4.46
			500	1.66			225	252			
	10		2000	3.34	2.44	105	222	272	288	118	4.46
			3000	3.94			236	241			
H. Moore	5	Mild steel	400	2.076	2.34	73	113	124	120	79	4.20
			500	2.276			115	127			
			600	2.421			121	130			
			700	2.627			119	127			
			800	2.775			121	128			
			900	2.928			122	127			
			1000	3.050			122	129			
			1100	3.163			123	130			
			1200	3.285			123	129			
			1300	3.435			123	127			
			1500	3.629			122	129			
			1700	3.826			121	129			
			2000	4.030			125	131			
	10	Mild steel	500	2.498	2.33	59	100	127	120	79	4.10
			800	3.040			107	129			
			1000	3.382			108	126			
			1500	4.028			113	126			
			2000	4.515			118	129			
			2500	5.013			119	127			
			3000	5.402			121	127			
			3500	5.795			120	127			
			4000	6.266			118	122			
	15	Mild steel	500	2.605	2.31	53	92	131	120	78	4.03
			1000	3.557			98	128			
			1500	4.212			106	130			
			2000	4.809			106	127			
			3000	5.745			102	126			
			4000	6.466			115	128			
Baker & Russell	10	Mild steel 880° C.	500	2.35	2.29	70	114	144	136	88	4.14
			1000	3.20			121	143			
			2000	4.32			130	141			
			3000	5.14			134	148			
		Stressed 19.77 tons	500	2.29	2.24	80	119	146	139	87	4.15
			1000	3.12			128	146			
			2000	4.23			133	150			
			3000	5.11			136	146			
		Stressed 23.36 tons	500	2.17	2.19	92	134	157	147	89	4.15
			1000	3.00			138	156			
			2000	4.10			145	157			
			3000	4.93			147	158			
		Stressed 27.15 tons	500	2.12	2.15	99	140	156	150	88	4.14
			1000	2.95			143	158			
			2000	4.07			147	158			
			3000	4.88			150	160			
		Stressed 33.04 tons	500	2.03	2.10	114	153	120	158	90	4.15
			1000	2.82			157	122			
			2000	3.94			157	121			
			3000	4.77			158	120			

gives this form of expression as applying to his data, whereas the constants he has supplied check his experimental Brinell numbers when applied to the formula $H = P^x \times C$. This was pointed out by O'Neill, who recognized it as the form of the initial term in his expression

$$H = yP^x + yP^x \times z/D$$

or the $H = KP^x$ that gives closely approximate values up to the maximum point.

Norbury (1923),[14] in tests upon annealed copper, used Meyer's formula, $P = ad^n$ and plotted, for different loads, $\log P$ vs. $\log a + n \log d$. He found, for recrystallized copper, that the straight lines intersected at one point where $\log d = 1$, that is, where the diameter of the impression is equal to the diameter of the ball. He found that $\log a + n$ was constant for each of the five specimens tested and claimed that it gave one value for the hardness of copper, independently of the previous mechanical or thermal treatment. This corresponds to the H_u of O'Neill.

Application of Various Formulas to Data of Different Observers

Applying these various formulas to miscellaneous data of different observers gives some indication of their reliability. Table 1 supplies ground for a few appraisals.

Thomson's equation is not general, derived for steel with a 10-mm. ball, and holds none too closely for the conditions of its application. (Figures are not given.)

Moore's number consistently gives a constant hardness number with varying loads and different ball diameters.

Meyer's copper and Moore's mild steel well illustrate the increase of the Brinell number to a maximum before a decrease with the increase of load. It is also evident that the $H_{max.}$ of Waizenegger and O'Neill gives a close value to the maximum indicated in the test, although it is not so consistent in value as Moore's number. O'Neill's H_u does appear to be unaffected either by D or the stressed condition of the material.

Norbury's $\log a + n$ gives excellent results, particularly for Baker and Russell's[15] stressed steels. H_u also holds very well for these steels, but not so well for Batson's steels. Norbury's number does not cover a sufficiently large scale for different degrees of hardness.

All the methods derived from Meyer's basic relation involve an additional source of error in the correct determination of the value of n, and also additional work and time in the computation of the hardness number.

If, after the initial load, the Brinell number is independent of the load, as found by Devries[16] and Batson,[17] the most obvious way is to measure the depth of the indentation. This, however, is not so easily done and requires the use of an additional instrument—a sensitive depth gage. If depth can be reliably measured by the indenting tool itself, best results will be obtained by applying an initial load to which a return is made before measuring the depth of penetration. This would be more accurate than measuring an indistinct diameter and would eliminate the effect of the varying load.

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Principles of Metallurgy of Ferrous Metals for Mechanical Engineers[1]

V—Cast Irons

By LEON CAMMEN,[2] NEW YORK, N. Y.

THE term "cast iron" has two widely different meanings. When an engineer specifies that the foundation in a machine will consist of two cast-iron bars he clearly means gray cast iron, which is by far the most widely used type of cast iron in commerce and engineering; but when a metallurgist speaks of cast iron he means the whole series of metals beginning with the softest gray iron and going through to the hard white iron, and including such widely different materials as plain gray iron, chilled iron, malleable iron, pearlitic iron, and the whole family of heat-treated and alloyed irons. It is because of this that the plural, "Cast Irons," is employed in the title of this chapter. Moreover, the conception of cast iron in all its forms is rapidly undergoing an important modification. Up to a few years ago probably the only two types of cast iron known were the metal as it came from the mold, which included gray iron, mottled iron, and chilled iron, all of which were brittle, and malleable iron, which was practically the only commercially used type of heat-treated cast iron and which had a certain amount of malleability, a tensile strength from 50 to 60 per cent superior to ordinary gray iron, and a certain amount of elongation.

Within the last ten or twelve years, because of a better understanding of the metallurgical principles underlying the production of iron-carbon alloys, a whole new series of cast irons have been created, some of them possessing higher ductility and tensile strength than the old, untreated, cast irons, and some endowed with properties which no alloy of equal carbon content ever possessed before. There has also been a tremendous improvement in the methods of manufacture of malleable iron which has resulted in the production of what may be considered to be a new material. Because of this, the subject of cast irons acquires a very great interest for the mechanical engineer, as within recent years this family of iron alloys has placed at his disposal many new and useful products, at times fully capable of replacing the more expensive steel castings or forgings.

The following incident may be of interest in showing how things have changed in this respect in American industry. When Mr. John N. Willys some twenty years ago took over what was then the American Motors Co., and has since developed into the Willys-Overland Co., the company had a large number of orders for automobiles on hand, but could not get them out because of the great delays in deliveries of malleable-iron castings and excessive rejections of the material when it was delivered. The situation finally reached such a state that manganese-bronze castings costing many times more per pound than malleable castings were substituted, and it was only then that a steady flow of automobiles began from the factory. Today hundreds of thousands of pounds of malleable castings are delivered of uniform quality and of strength far superior to the best that was available twenty years ago.

Looking over the whole series of alloys of carbon and iron beginning with the softest iron and ending with the cast-iron group, it would appear that as the carbon content increases from zero to 0.89 per cent there is a gradual replacement of the original ferrite by pearlite, till at the eutectic limit the pearlite forms the whole mass of the metal. With the further increase of carbon some of this pearlite is replaced by free cementite till at 1.70 per cent of carbon (the region to the left of point *E*, Fig. 5, MECHANICAL ENGINEERING, May, 1925, p. 345) the mass consists of pearlite (86 per cent by weight) and free cementite in the ratio of 6.1 of pearlite to one of free cementite.

As we pass beyond the 1.70 per cent carbon we run into the possibility of formation of two groups, which may be either gray irons or white irons. The increase of the carbon content[32] from 1.70 to 4.30 per cent is accompanied by a continuous replacement of the structure which existed at that time (Howe calls it "1.70 per cent austenoid," in view of the fact that it is a product of the transformation in cooling of the austenite of 1.70 per cent carbon which forms in the solidification of all alloys containing more than 1.70 per cent of carbon) by a eutectic, till at 4.30 per cent of carbon the eutectic forms the whole mass. With further increase of carbon

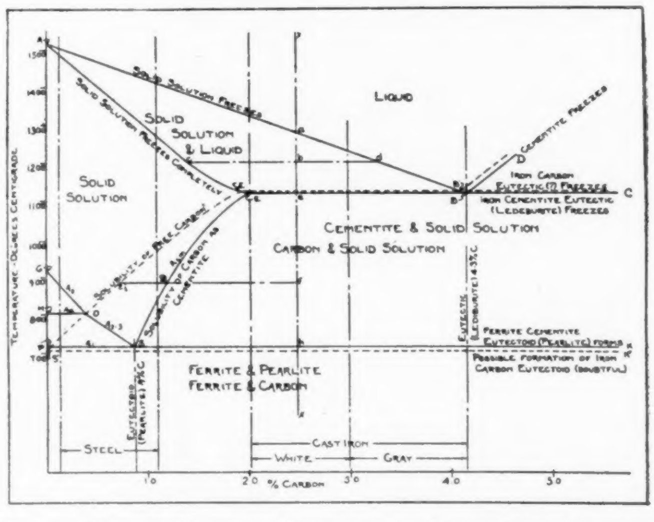

FIG. 14 BENEDICKS' DIAGRAM RECORDING THE EQUILIBRIUM CONDITIONS IN TERMS OF TEMPERATURES AND AGRAPHIC (NON-GRAPHITIC) CARBON
(From American Malleable Cast Iron, H. A. Schwartz, Penton Pub. Co., 1922, p. 47. Reproduced by courtesy of the publishers.)

content this eutectic is in turn replaced by cementite, till at 6.67 per cent of carbon the mass would theoretically consist of cementite alone, but for certain reasons never does.

There have been a good many attempts to define cast irons in a manner to separate them entirely from high-carbon steels into which they can easily pass. However, most of the definitions given hitherto are not suitable for the state of things that exists today. H. M. Howe[1] states, "Cast iron usually contains so much carbon that it is not usefully malleable at any temperature." But the late Professor Howe's former associate, Prof. Bradley Stoughton,[2] describes a cast iron which can be "tied into knots." Better definitions can probably be obtained by going back to the constitutional diagram of the iron-carbon system (Fig. 5, p. 345, MECHANICAL ENGINEERING, May, 1925). It was pointed out there that as we move from left to right we pass gradually from hypoeutectoid steels consisting of ferrite and pearlite through the stage of purely pearlitic (eutectoid) steels into hypereutectoid steels consisting at room temperature of cementite and pearlite, the amount of cementite increasing with the carbon content. This constitutes the region lying to the left of *E*. To the right of *E* begins another region which may be classified as that of cast irons. The slowly cooled alloys of concentration lying between the points *E* and *B* should consist at temperatures below *PK* of crystals of cementite imbedded in pearlite, provided the cooling took place at a certain moderate rate. If this did not happen, then the structure would be one containing pearlite and products of the degeneration of cementite, with or without the presence of ungenerated cementite.

[1] Fifth of a series of articles discussing the underlying physical and chemical processes involved in the metallurgy of iron and steel. The first, dealing with the physico-chemical properties of iron alloys, appeared in the May issue, p. 339, the second, on the crystalline structure of ferrous metals, in the June issue, p. 479, the third, on determination of the properties of metals, in the July issue, p. 559, and the fourth on heat treatment, in the August issue, p. 637.

[2] Associate Editor, MECHANICAL ENGINEERING.

From this point of view the diagram of Fig. 14 is of great interest.[3] In this diagram the abscissas, represent carbon content in per cent and the ordinates, temperatures in degrees centigrade. The conditions of equilibrium are then represented by various lines and fields on the diagram. Thus above *ABD* the metal is a homogeneous liquid. *ABD* shows how the carbon content affects the beginning of freezing, while *AEBC* does the same for the completion of freezing. In the areas *AEB* and *DBC* the metal consists of a mixture of a homogeneous liquid and (theoretically) a homogeneous solid, which, as has been shown elsewhere, is a "solid solution." The alloy of lowest freezing point *B*, carbon 4.3 per cent, is known as the eutectic. The solidification of non-eutectiferous

Fig. 15 Eutectic Cast Iron Rapidly Quenched in Water Showing Ledeburite (Wüst)
(From Cast Iron in the Light of Recent Research, by W. H. Hatfield, Charles Griffin and Co., London, 1918.)

alloys to the left of *E* is completed by the freezing of a solid solution of composition dependent on the original carbon content at temperatures marked by the line *AE*. Below the freezing point other rearrangements occur in the solid metal.

The mechanism of formation of cast irons is not entirely certain as yet. It would appear, however, that the molten mass represents a mixture, probably a solution of carbide of iron in iron. The line *BC* of Fig. 5 (Mechanical Engineering, May, 1925, p. 345) represents probably the temperatures at which the carbide and molten iron are in equilibrium. A further cooling results in the precipitation of free carbide and it is only after the resolution of the eutectic at about 1130 deg. cent. or line *ED*, into free carbide and austenite that the free carbon is formed. The degree of rapidity with which the cooling of the iron both before and after freezing takes place is the factor which mainly determines the ratio of free to combined carbon in the finished product, assuming the same chemical composition. Wüst[4] investigated the composition of the eutectic point and its temperature of solidification as well as the microstructure of this eutectic alloy. Fig. 15 shows a sample of cast iron of eutectic composition quenched in water and magnified 100 times. The structure presented by the solidified eutectic is clearly distinct from any of the steel structures such as shown, for example, on page 343 of Mechanical Engineering, May, 1925, and Wüst proposed to call it "ledeburite." From these tests it would appear that the production of the graphite takes place progressively during the recalescence observed at about 1130 deg. cent. (line *ED* or *BD*).

Various Kinds of Cast Irons

Cast iron, using the term in the broad sense, is so affected by its composition, process of chilling, and, at times, subsequent heat treatment that a variety of materials is obtained possessing a wide divergence of properties. Cast iron may be made soft enough to be perfectly machinable and also glass-hard; very brittle, and so tough that it can be tied in knots, and may have a range of tensile strength from about 18,000 lb. to 55,000 lb. per sq. in. The following is an enumeration rather than a classification of the types of cast iron of special interest to mechanical engineers. It does not claim to be logical but it is believed that it will be of practical interest.

Direct Castings. In a broad sense the product of the blast furnace is cast iron. It is not usually so called, because in the vast majority of cases it is poured into pigs or, after passing through the mixer, is delivered to bessemer converters or open-hearth furnaces. Many efforts have been made, however, to produce castings directly from the material delivered by the blast furnace. In particular it was stated in the press a couple of years ago that the Ford Motor Co. was working on this process. Such castings are called "direct castings." Hitherto no special success has been attained, except in a few cases where very large castings were made.

Gray Iron. The characteristic feature of gray-iron castings is that more or less of their carbon is present as graphite, while in white iron, as will be seen later, it is present as cementite, and in malleable iron as "temper" carbon. (The meaning of this latter term will be explained in connection with the production of malleable castings.) This graphitization is usually far from complete, though in some exceptional cases, as in "black heart" malleable castings, it may be practically complete. When a mechanical engineer speaks of "cast iron" without specifying it further, he usually means gray iron, and the author has received a letter from Dr. Richard Moldenke, Chairman of the Committee on Cast Iron of the American Society for Testing Materials, stating that whenever cast iron is referred to in the specifications of that society without further explanation, gray cast iron is meant thereby. Gray cast iron is soft and easily machinable, except for the outside skin of the castings, which may have sand from the mold burned into it and therefore be very hard on the tool, producing a kind of abrasive action. As appears from Fig. 16, the carbon is in the form of flakes. The presence of these flakes does not directly affect the strength of the iron matrix, but acts as that much hollow space, reducing the amount of the material over the cross-section capable of resisting mechanical stresses. Because of this gray cast iron has a comparatively low tensile strength, varying usually from 18,000 lb. and less per sq. in. up to 22,000 or 23,000 lb. Furthermore, the shape of the cavities in the iron matrix produced by the carbon flakes, as appears from the same illustration, is extremely irregular, abounding in sharply bounded promontories. Each one of these acts in essentially the same manner as a notch in the notch impact test and reduces at numerous points distributed throughout the material the ability of the latter to withstand shock, making gray cast iron brittle.

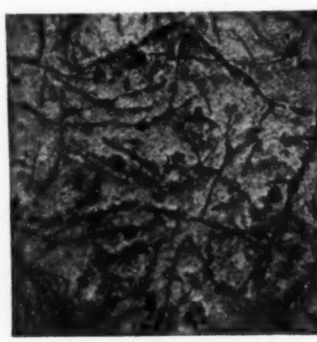

Fig. 16 Microstructure of Gray Cast Iron as Cast
(Black lines represent graphite plates.)
(From Cast Iron in the Light of Recent Research, by W. H. Hatfield, Charles Griffin and Co., London, 1918.)

White Cast Iron. Carbon in cast irons may occur essentially in one of three forms: as free carbon, in one of two main forms, viz., as graphite flakes or plates (in gray cast iron), or as rounded globules, as in malleable iron described more fully below. It may also occur combined with iron, i.e., as cementite. This happens in white cast iron (Fig. 20), to which cementite imparts its properties of hardness and brittleness. Because of the combination of these properties, white cast iron has few industrial uses.

Mottled Cast Iron is an iron which, because of the conditions of cooling, is an intermediary product between white and gray cast iron, and approaches the properties of either one of the two depending upon which of the two types is prevalent in its structure. The alternation of white and gray spots gives it the mottled appearance which led to the adoption of its name.

Chilled Iron. When pig iron cools from the molten state it holds in solution, probably in the form of carbide of iron, all the carbon which it contains at a given temperature. On cooling, some of this carbon separates out in the form of graphite flakes which are distributed throughout the mass. The remainder stays in the

form of combined carbon (cementite), the effect of which is to increase the hardness of the metal. The separation of the graphite, as shown elsewhere in this article, depends mainly upon the rate of cooling, though also upon the silicon content of the metal, so that, all other things being equal, if the iron is cooled very suddenly and is of proper composition, all carbon may be retained in solution as combined carbon, which renders the iron dense, white, intensely hard, and capable of receiving a very high polish. Such an iron is called "chilled iron." It is used for making railroad wheels, rolls for shaping steel, etc. In making rolls only the exterior layer is given a chill, both because it would be extremely difficult to chill the whole casting quickly enough to produce the required structure throughout, and also because there is an advantage in having the roll possess an extremely hard surface (to reduce the wear against the metal rolled) and at the same time a soft interior to cushion the shocks and thus reduce the breakage of the rolls. This chilling of only a part of the roll is effected by making that part of the mold corresponding to the necks and wobblers of sand, while that part destined to form the body is made up of a heavy cast-iron ring. The rapid cooling caused by the absorption of the heat by the cold ring casting in contact with the molten metal causes a chill on the outer surface of the roll, the depth and hardness of which is controlled by varying the composition of the molten metal.

The following analysis of different parts of a chilled roll is given in The Making, Shaping and Treating of Steel, by J. M. Camp and C. B. Francis, 2d edition, p. 326:

	Total carbon	Combined carbon	Graphite carbon	Silicon	Sulphur	Phosphorus	Manganese
Chill	3.00	3.00	...	0.90	0.04	0.200	0.25
Mottled	3.00	2.25	0.75	0.90	0.04	0.200	0.25
Gray	3.00	1.00	2.00	0.90	0.04	0.200	0.25

*DeLavaud Metal.** Centrifugally cast pipe has become an important product both in this country and abroad. It is made by several processes, one of which is the DeLavaud process. In this a rapidly rotating, steel, water-cooled mold is used, the metal being delivered from the end of a horizontal spout. The spout is stationary while the mold has not only a rotary movement but moves at a certain predetermined speed away from the spout, so that the stream of metal coming from the spout impinges upon the mold and forms something like a ribbon longitudinally wound with overlapping edges. The metal as cast is hard and somewhat brittle. It is, however, passed through an annealing furnace where it is heated for a few minutes to a temperature of the order of 1750 deg. fahr. (955 deg. cent.).

The following information as to the structural condition of the annealed DeLavaud pipe is taken from A Study of DeLavaud Centrifugally Cast Cast Iron, by Arthur Phillips, copyright 1923, U. S. Cast Iron Pipe and Foundry Co., pp. 11–12. According to Mr. Phillips, DeLavaud pipe is made up of three gradually merging layers, namely, an outside layer similar to the structure of malleable cast iron, an inner layer consisting of a steel matrix (containing impurities, of course), the continuity of which is taken up to some extent by graphite, and an innermost layer of gray cast iron. The high tensile strength of the DeLavaud pipe, which tests have shown to be of the order of 32,000 lb. per sq. in. average with an elongation of 0.5 per cent in 2 in., is probably due to this unusual combination and arrangement of structural elements, as the chief weakening and embrittling element in gray cast iron is the soft and friable graphite, and in the DeLavaud pipe the center of the wall consists of a steel matrix containing graphite in small areas which do not break up the continuity of the mass as much as the usual irregularly shaped flakes.

The Wasson Piston Ring Co.[5] attempted to use the DeLavaud method for casting piston rings. The castings came absolutely hard and were annealed in the same way as is done with pipe. The annealing process, however, precipitated the carbon out of the combined state into temper carbon and not graphitic carbon. The result was that when the rings were put into a test aircraft engine, the latter started off with the highest efficiency that it had ever shown, but the efficiency very soon fell off and at the end of 54 hours the rings had lost twenty thousandths of an inch on the wall and the cylinders were cut all to pieces. As the carbon was in the form of temper carbon, the material was not suitable as a bearing material.

Annealed Gray Cast Iron

Among the processes for annealing gray cast iron attention has been drawn to that of Alexander K. Schaap described by Prof. Bradley Stoughton.[6] In this process the casting is heated to a temperature slightly above the critical (1600 deg. fahr. = 871 deg. cent.), while protected in a muffle surrounded by a gas flame and open at the top. A good deal depends on the construction and material of the muffle for the success of the process, and, for example, a crack developed in the side of the muffle produces hard spots opposite the opening. When the iron reaches the proper temperature the muffle and its contents are removed from the furnace and allowed to cool in the open air, the casting being protected from drafts by laying a cover on top of the muffle. In this way it requires about fifteen minutes for the iron to cool to a black heat, after which it may be cooled in the open air.

Fig. 17 Annealed Cast Iron
(From an article by Prof. Bradley Stoughton, *Iron Age*, Jan. 3, 1924, by courtesy of the author.)

The material is said to have very unusual qualities, particularly as regards resiliency. A thin bar may be coiled up into a spiral and used as a spring, although, of course, it will not be nearly as strong and resilient as a tempered-steel spring. This iron does not "grow" by repeated heatings and coolings and does not revert to the white cast iron formed upon heating followed by rapid cooling.

The original structure of this metal as cast consists of coarse grains of graphite leaving only short and separated metallic paths of rupture. The treatment produces an almost continuous metallic phase with relatively small particles of carbon between. It would appear that in the course of treatment the following takes place: At a temperature of 1600 deg. fahr. or slightly above the coarse flakes of graphite are apparently absorbed to some extent in the solid solution existing at that temperature. With the moderately slow cooling which follows, the solid solution breaks up into ferrite and pearlite mixed with fine particles of carbon surrounded by crystals of ferrite from which it has separated. The pearlite

* This term has been borrowed from the Specifications for the DeLavaud Centrifugally Cast Cast Iron Pipe issued by the U. S. Cast Iron Pipe and Foundry Co., Burlington, N. J., the largest makers of this material in the United States.

matrix seems to be semi-globular in structure rather than the lamellar (plate-like) pearlite of annealed white cast iron. (Fig. 17.)

The structure of the iron indicates clearly the difference between this metal and malleable cast iron made by the usual annealing process of $6^1/_2$ days: The latter consists of a ferrite matrix with particles of temper carbon; the purpose of the annealing is to completely decompose the cementite of pearlite into carbon and ferrite. This doubtless explains the necessity for the long time of annealing for malleable cast iron, and also the lack of resiliency, which could not be expected in an almost continuous ferrite phase. On the other hand, the resiliency of this annealed iron, which so resembles that of steel, is apparently due to the pearlitic, instead of ferritic, phase.

Statements have been made that the same results can be achieved without resorting to the particular process (use of muffle) employed by Mr. Schaap.

Pearlitic Cast Iron.[7] This is one of the latest important developments in cast irons, but the information available about it is still somewhat sketchy. Pearlitic cast iron is an iron which is of such a composition that if cast in an ordinary mold it would give a white fracture. Through being cast, however, in a mold heated to a certain temperature either before or after casting and through having, therefore, the cooling of the metal retarded in a certain manner, a microstructure is produced which consists of a pearlite matrix with enclosures of free graphite and which therefore gives a metal of superior strength. The interesting part about pearlitic cast iron is that it represents a deliberate and successful attempt to create in a metal a structure which previous metallographical investigation has shown would contribute to the improvement of its physical qualities.

Pearlitic cast iron is an iron which has a low silicon and low total carbon content, the temperature of the mold depending on these two factors of composition and the thickness of section. Thus, for example, with a total carbon content of 3 per cent, including graphite 2.15 and silicon 1 per cent, the mold should be reheated to a temperature varying as follows with the thickness of the section of the casting: Thickness of section 0.4 in.—to about 200 deg. cent. (392 deg. fahr.); thickness of section about 0.8 in.—to about 150 deg. cent. (302 deg. fahr.); and thickness of section 1.2 in.—to about 100 deg. cent. (212 deg. fahr.). While the plans controlling this process (it was invented by Diefenthaler at the Lanz Foundry in Germany) specify the manganese and phosphorus contents, it does not appear that they attach any importance to them. The sulphur content does not appear to be of importance unless excessive.

Malleable Cast Iron. There are two products coming under this name. The first historically, though the lesser in importance industrially today, is the so-called "white heart" malleable iron invented by Réaumur in the first quarter of the eighteenth century and still made on a restricted scale in Europe. This iron is produced by packing small white-cast-iron castings in pulverized hematite ore or mill scale and heating them to bright redness for many days. The surface becomes decarburized by the contact relation between the cast iron and the packing. As practiced today, however, the heat treatment is insufficiently deep and the cooling of the piece excessively rapid. Because of this, part of the carbon is in the free state and part in a solid solution. The structure of the material lacks in uniformity and the central part of the casting, especially if it is comparatively thick, retains the normal structure of white iron. If the casting comprises sections of different thicknesses, some of them will be free of this white-iron core or heart and some will possess it, with the result that the pieces are of unequal strength and somewhat subject to warping and cracking.

The American method, invented toward the end of the first half of the nineteenth century by Seth Boyden in Newark, N. J., is worked in an entirely different way. As in the European process, the start is made with white iron, but this is annealed for a longer or shorter time at a temperature well above the upper critical point and then cooled extremely slowly at a rate not exceeding 10 or 11 deg. fahr. per hour as a maximum. The result is a material of great uniformity of structure, even in complicated shapes, and of a high tensile strength and high coefficient of elongation.

This has been materially assisted by the way manufacturers of malleable-iron castings have gone about their business in the United States, namely, by adopting the so-called "certified castings" scheme. The majority of manufacturers have formed an association which has appointed a laboratory as their testing plant. Sample bars from various heats are sent by each manufacturer to the central laboratory where they are tested, and only those manufacturers whose bars during a certain period have maintained a uniformly high standard are entitled to call their products "certified." When the bars do not live up to the desired standard, engineers are sent out from the central laboratory to investigate the manufacturing conditions at the plant. By this means of inspection and coöperation the average standard has been raised and some of the manufacturers have been enabled to attain a remarkably high state of excellence of their products.

Most of the malleable iron made by the better manufacturers today ranges from 2.30 to 2.70 per cent carbon before annealing.[8] The size of the casting affects the rate of cooling, and thereby the tendency to graphitize on freezing. This tendency can be controlled by varying the silicon content. With little or no silicon, graphitization in the annealing process is retarded to a commercially prohibitive degree. When too much silicon is present, graphitization may take place during freezing, which is undesirable. Because of this the silicon content varies from 0.60 to 0.80 per cent, the general practice being to vary the silicon inversely with the carbon, and for a given carbon inversely with the cross-section of the casting.

As cast, white iron consists essentially of cementite embedded in pearlite or a solid solution of iron. When this is raised to a temperature well above the Ac_1 point, say, 1700 deg. fahr., a decrease in cementite and an increase in the formation of solid solution takes place.

As the heating at 1700 deg. is continued the cementite breaks down and produces free carbon. While, however, the free carbon in gray iron is in the form of irregular flakes, the free carbon produced from cementite under the conditions here described appears in the form of fine globules distributed through the metal mass with fair regularity. Such carbon is called "temper" carbon. The structure then consists of temper carbon and a solid solution of carbon or carbide of iron in iron (there is still some doubt which is correct), and the product is still whitish in fracture and very brittle. If, now, the temperature is carried down to 1300 deg., (i.e., below the Ar_1 point), or slightly below that, complete graphitization of the product takes place. A good deal depends on the duration of the first heating and subsequent cooling. Commercial practice in America involves a maximum temperature of the casting for a period from 24 to 60 hr. and an average rate of cooling from this not faster than 10 deg. per hour, preferably less, particularly near the critical point.

An interesting example of what may be considered as the most modern type of annealing plant will be found in the installation of the International Harvester Co., described in *Foundry*, vol. 52, no. 2, Jan. 15, 1924, pp. 43–50. Among other things accomplished there, the time of anneal was cut by the use of a furnace of the tunnel type. The average tensile strength of the product is given at 52,048 lb. per sq. in. with an elongation of 13.80 per cent in 2 in.

The necessity of holding the iron in the first heating stage and subsequently in the cooling stage for a time varying from 5 to 9 days naturally enormously increases the cost of the product, both on account of the fuel consumption and on account of the low factor of production per unit cost of the plant. There have been numerous attempts to produce malleable iron by means of only a very short anneal.[9] In one case the white iron contained 2.23 total carbon and 0.94 silicon. The material was heated for three hours at 1700 deg. fahr., then for one hour at 1560 deg. fahr., and was cooled uniformly to 1320 deg. fahr. for 14 hours and then to 1200 deg. for 12 hours, and it is believed that an even more rapid treatment is possible. It is claimed that a material was produced with tensile properties varying from those of malleable iron to values of about 70,000 lb. per sq. in. strength and 4 per cent elongation, and that even higher values were obtained. This product is not strictly a malleable iron, though it may take its place. As a matter of fact, attempts to achieve such results are not at all new, and as early as 1902 Alex. E. Outerbridge, Jr., described his work of very much earlier date,[10] in the course of which white iron was converted into a machinable material by a short anneal at a very high

temperature. The possibility of this as a means of commercially producing standard-specification malleable-iron castings is, however, vigorously denied by some men in the trade.[11] In general,

> ...speed is promoted by graphitizing the cementite at the highest possible temperature, but to a certain extent at the expense of quality. Temper carbon differs from graphite only in form. It has been pointed out that these differences of geometric form are due to the temperature of the metal in which the free carbon is formed. Accordingly, the two forms, temper and graphitic, shade over into each other by infinitesimal degrees, and the temper carbon formed at high temperatures may grow so coarse and flaky as to be almost graphitic.[12]

An interesting application of malleable-iron castings is in the manufacture of heat-treated malleables, which may be used, for example, for tools. The heat treatment depends upon the usage which the casting is to undergo. One of these treatments used for the production of hammer heads consists in heating the hammer head to about 925 deg. cent. (1697 deg. fahr.) and dipping it in warm water two or three times, with intervals of about one second between the quenchings. Among other things, twist drills are produced which work such materials as brass, gray iron, malleable iron, steel, and even hard white iron. The latter is particularly interesting, as usually only high-speed steel can handle it.

Synthetic Cast Iron.[13] Synthetic cast iron is nothing but steel scrap, particularly turnings carburized by melting these materials in the presence of carbon which is introduced simultaneously with them in the melting appliance, the electric furnace being particularly suitable for this purpose.

Its somewhat extensive use was brought about by conditions prevailing during the World War. Then enormous amounts of steel turnings and small scrap became available as a result of the manufacture of munitions. Such turnings can be used in the open-hearth furnace only in briquetted form, as otherwise they are extremely difficult to charge and oxidize so rapidly before melting as to produce enormous losses. The production of the turnings at that time proceeded, however, at such an enormous rate as to exceed all possibility of their being briquetted with the facilities then available. In looking around for a use for them the production of synthetic cast iron was resorted to. It is rather unlikely that it will be carried on under normal peace conditions, except where the amount of small steel scrap and turnings is very large, means for producing cast iron in a regular way are lacking, and electric power is available at a cheap rate. Such conditions may, for example, arrive in some of the South American countries, and perhaps in regions like Texas.

Semi-Steel. Essentially semi-steel is a metal made up of a mixture of cast iron and steel, a mixture which may be produced in several ways. Thus, molten steel may be added to molten cast iron in the ladle, or steel in the form of scrap may be charged into the cupola. The latter is the standard method. The best way to make semi-steel appears to be to use a hematite low-phosphorus iron, although in Europe very high-phosphorus iron (1.3 to 1.4 per cent phosphorus) has been extensively used. It is very important to have the manganese in the mixture high, and the best semi-steel for general work should contain 0.75 to 1 per cent of manganese. The best steel to be used for this purpose is mild steel, especially when in the form of plate cuttings and clippings from $^1/_2$ to $1^1/_2$ in. in thickness and in moderate-sized pieces. Crop ends of rails and similar material have also been found suitable, but not high-carbon steel. The percentage of steel used has varied from 15 to 40 per cent, the lower figure for light work and the higher proportions for heavier castings. There is still a good deal of uncertainty as to the methods of its manufacture and processes which have failed in one foundry have been found to be very successful in another, without any clear reason why.

During the war large amounts of semi-steel were used for the manufacture of shells, aerial bombs, etc. and in many cases gave excellent results. The average analysis of good semi-steel, for example, in British aerial bombs, is as follows:

Silicon	2.13	Graphitic carbon	2.55
Phosphorus	0.065	Manganese	0.624
Combined carbon	0.520		

The tensile strength of the material is well in excess of 40,000 lb. per sq. in. In general, it has been found that to consistently maintain a sufficiently high standard in semi-steel a foundry should produce substantial quantities daily, and it is very difficult to produce good material on a small scale or by incidental work. It has been found that when semi-steel is annealed at a temperature from 760 deg. to 800 deg. it becomes very soft, with a certain amount of malleability and ductility, but loses from a quarter to one-third of its tensile strength.

Annealing at a lower temperature has been tried with a much smaller loss in tensile strength.[14,15]

Alloying Elements in Cast Irons

It is legitimate to consider all the elements in cast irons except iron and carbon as alloying elements. Some of these, as sulphur and phosphorus, may be also regarded in the light of impurities, as they mainly come from the ores or fuel, and the metallurgist has to keep them down to a limit rather than, except in some cases, deliberately introduce them. Other elements, such as nickel, are introduced deliberately, in order to achieve a certain purpose such as to increase the strength of the material. The information available as to the influence of the alloying elements in cast irons

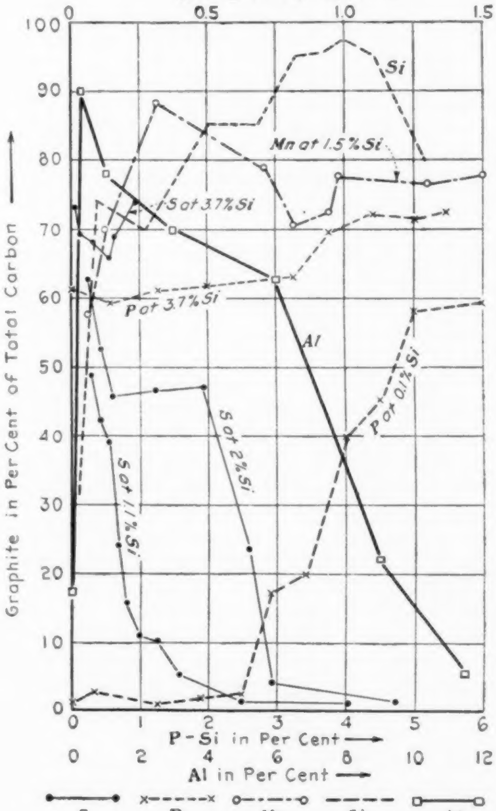

Fig. 18 Influence of Some of the Elements Present in Cast Iron on the Formation of Graphite
(Based chiefly on results obtained by Wüst and his collaborators.)

is still somewhat sketchy, particularly with respect to the rare metals.

Silicon. Next to carbon, silicon is the constituent of cast iron capable of exerting the greatest influence on the structure of the material.

In general, it has been found that as the silicon is increased the carbon changes from the combined to the graphitic state. It is important to remember, however, that this influence may be either assisted or counteracted by other factors, the most important of which are the rate of cooling, the size of the casting, and the casting temperature. The size of the casting in itself is important only because, all other things being equal, it determines the rate of cooling, a thin section obviously cooling faster than a thick, heavy one. Because of this, more combined carbon will be found in a thin section than in a thick section (assuming the same silicon content in both). The rate of cooling is important because the faster the cooling, the more will there be tendency toward the formation of

combined carbon. The casting temperature is also important. It has been found that the higher the casting temperature, the more tendency there is toward holding the carbon in the combined form. W. H. Hatfield[16] carried out some tests in the course of which he cast two bars 2 in. × 1 in. × 12 in. from the same ladle of iron, the first bar being cast at 180 deg. cent. higher temperature than the other. Upon breaking the bars the higher-temperature one was found to be mottled white, while the low-temperature one was found to be gray. The microstructure in the white areas of the first corresponded with the normal white-iron cementite-pearlite structure, while that cast at the lower temperature upon examination proved to have a microstructure substantially approaching that of regular gray iron.

In this connection Fig. 18 (reproduced from p. 91 of Das Technische Eisen, by Dr. Paul Oberhoffer: Julius Springer, Berlin, 1925) is of interest as it shows the influence of the various constituents of cast iron upon the formation of graphite. In this figure the graphite is expressed in percentages of the total carbon, and it would appear therefrom that silicon up to about 4 per cent increases the formation of graphite. When, however, present in excess of it, it evidently begins to act the opposite way. It would appear also that there is a certain irregularity in its action between 0.3 and 1.1 per cent.

Sulphur. The actual influence of varying proportions of sulphur upon the physical properties of cast iron is not yet clear. It would appear, however, that in the absence of manganese it assists in retaining the carbon as a carbide, thus producing hard iron. If the iron is very high in manganese, manganese sulphide is formed and the hardening action is neutralized, but other effects are produced.

It does not appear that small percentages, say, up to 0.08, affect the mechanical properties of cast iron seriously, and whatever influence it has is largely concealed by such other factors as temperature, cooling, etc. It would also appear that the hardness of the iron is not substantially affected by low sulphur, while its ability to bend and its tensile strength increase with sulphur up to 0.1 per cent. When, however, sulphur content is high, hard spots develop and the iron becomes brittle. High percentages of sulphur are often encountered in iron which has been much remelted or in iron in which a large percentage of scrap, possibly remelted before, is employed. Lately a process has been developed for lowering the sulphur content in cast iron by a certain use of soda ash.

Manganese. While it is known that manganese has an important effect on the properties of cast iron, its actual influence is still not quite understood. If manganese and sulphur are both present, a compound manganese sulphide is formed which does not affect the properties of the carbon. In steel the presence of manganese is often undesirable. In rolled products it is apt to roll out into a thread or streak in the direction of the rolling, which does not materially affect the strength of the metal in that direction but seriously reduces it in the direction transverse to rolling. In malleable cast iron a similar effect is possible. In itself, manganese raises the degree of solubility of carbon in iron.

Phosphorus. The influence of phosphorus has been investigated by Dr. A. E. Stead,[17] from whose observations it would appear that iron and phosphorus alone form mainly a solid solution of phosphide of iron Fe_3P in iron, reaching eutectic composition at 61 per cent of phosphide of iron, and 39 per cent of the saturated solution of phosphide of iron in iron. The effect of carbon is to throw out free phosphide in increasing proportions with each increment of carbon. Apparently there is a competition between carbide of iron and phosphide of iron for iron with which to form a solid solution and carbide of iron prevails. When the carbon has passed the two per cent limit there appears to be evidence of a carbide-phosphide eutectic containing about 6.7 per cent of phosphorus and 2 per cent of carbon. Since both the phosphorus and carbon need a certain quota of iron to be held in solution, it would appear that the presence of either will decrease the amount of the other which can be present in iron. Indeed, Stead found that 1.56 per cent of phosphorus appears to be responsible for lowering the carbon content from 4.10 to 3.75 per cent.

Phosphorus also appears to have an influence on the condition of the carbon in that it increases the ratio of graphite to combined carbon. This has not yet been completely proved, however.

As regards physical properties, it would appear that phosphorus when present in quantities of from 0.3 to 0.6 per cent increases the tensile and bending strength of the iron, but reduces the tensile strength when present in greater quantities. On the other hand, hardness increases steadily with phosphorus content.

A foundryman, especially when making intricate castings, likes to have the phosphorus in his iron high, because of the low melting point of such iron, which reduces the fuel consumption in melting and permits pouring the metal at a lower temperature. The iron, however, is less strong and more brittle, the latter being due to the fact that free phosphide of iron is present and not only does not add anything to the strength of the metal (in this respect it is similar to the flakes of graphite), but is likely to create planes of weakness owing to the differences in the coefficients of expansion between the iron matrix and the phosphide inclusions.

The presence of phosphide of iron is easily detected by the heat-tinting process[18] (Mechanical Engineering, July, 1925, p. 561), as carbide of iron on heating takes the oxidation tints of straw yellow, yellow, brown, red, purple, blue, and silver gray in sequential order. Phosphide of iron passes through the several ranges of color, but not so rapidly as the carbide, so that when the carbide shows under the microscope as red, the phosphide is a pale yellow, and when the carbide has been heated to blue tint, the phosphide will be brown or red-salmon in color.

Nickel. The presence of nickel in cast iron[19] causes increased graphitic-carbon formation, i.e., nickel tends, as does silicon, to "gray" the iron. This effect is quite definite but very mild; one per cent of nickel being perhaps equal roughly to from one-quarter to one per cent of silicon in this respect.

The nickel does not form in the carbides, but dissolves in the ferrite of the iron. If the composition is such that combined carbon is present in the castings, this will be fine in structure in the presence of from 1 to 5 per cent nickel, resembling sorbite more than pearlite and being in consequence harder.

Therefore it will be seen that nickel exercises two effects quite opposite in nature; by lowering the combined carbon it softens the iron, but by sorbitizing the pearlite matrix of the iron, it hardens it. Which of these will prevail depends largely on the amount of combined carbon. If this is low in the composition of the iron under consideration, i.e., if the iron is soft and open, the nickel will soften it still further. If it is high, from 0.3 to 0.8 per cent, the hardening effect will predominate, as in fact it generally does in practice.

Grades of gray iron carrying from 0.50 to 0.85 per cent of combined carbon in general will be hardened, strengthened, and toughened by the addition of from 1 to 5 per cent of nickel. Increases of from 15 to 40 per cent in hardness and in bending or compressive strength are thus obtained. The use of nickel is often beneficial, particularly in thin sections, in that good values of hardness may be obtained without incurring the risk of obtaining chilled or mottled iron or hard spots. The hardness obtained in gray iron by nickel additions is not due to an increase in the amount of carbide present as it is when obtained by lowering the silicon content, with consequent attendant machining difficulties, but is due to the superior and finer form of the pearlite or carbide in the presence of nickel.

The effect of the addition of from 1 to 5 per cent of nickel on the fluidity and the shrinkage of cast iron appears to be so slight as to be inappreciable. However, the addition of nickel appears to produce finer-grained castings, particularly in thin sections. The resistance to corrosion of nickel-bearing cast iron is somewhat superior to that of similar iron without nickel, but probably not sufficient to warrant the use of nickel generally to increase corrosion resistance. On the other hand, the resistance to oxidation at high temperatures is increased by the addition of nickel, and some efforts have been made to utilize this feature in a practical manner.

When nickel is used in connection with chromium, the combined effect of the two elements depends upon their relative amounts, but generally is in the direction of hardening and strengthening the iron and increasing its chilling power.

Nickel cast iron is used for cylinder blocks and pistons of automobile engines by the Cadillac Motor Car Co. among others. From service tests of cylinders this company has found that with the lower hardness a wear of from 0.0015 in. to 0.002 in. was obtained in 20,000 to 25,000 miles and the cylinders would need re-

grinding. With a Brinell hardness of 175 to 200, however, the cylinders showed a wear of only from 0.00075 in. to 0.001 in. and did not need regrinding. The company credits this increased wear not only to the hardness of the iron, but also to the fine grain and high-luster finish which can be produced with addition of nickel. It has been found that the nickel-cast-iron cylinders at 200 Brinell hardness machine as easily as the nickel-free ones at 150 Brinell hardness.

Mayari Iron. This is an iron made from ores (Cuba) naturally containing nickel and chromium, the former 0.80 to 1.25 and the latter from 1.60 to 2.50 per cent, together with small percentages of titanium and vanadium. The silicon and carbon may be modified within considerable ranges.

It is stated[20] that in this iron the nickel and chromium work to a certain extent in opposite directions, the former tending to separate out the graphite and the latter to hold the carbon in the combined form. It is claimed, however, that Mayari iron gives castings increased strength and solidity. It is used specifically for wear-resisting castings, such as rolls, jaws, and parts for rock crushers, balls and liner plates for ball mills, rubber-mill rolls, etc. and has been extensively employed in the manufacture of rolls for metal rolling, both chill and sand types.

Cobalt. O. Bauer and E. Piwowarsky[21] state that cobalt retards graphite precipitation and favors the formation of carbides, its action being therefore exactly the opposite of that of nickel. When present in cast iron it rapidly reduces the ability of the latter to bend, and slowly reduces the tensile and compressive strength of the metal; it slightly increases the hardness.

Titanium.[22,23] Titanium in cast iron acts essentially in the same way with respect to graphitization as silicon, but more powerfully, so that independently of the silicon content the maximum of graphitization is reached at 0.1 per cent of the total titanium content. Until this maximum amount is reached, the influence of titanium in the direction of graphitization surpasses every other influence in so far as the mechanical properties of cast iron are affected. If more titanium is added the graphitization proper is not affected but the mechanical properties of the metal are noticeably improved, and, for example, bending is improved 50 per cent or better.

Uranium.[24] The most important features of uranium as an addition to cast iron are said to be its effectiveness as a scavenger and its influence on the size and distribution of the graphite in the iron. Microphotographic investigation and chemical analysis show that while no change in the composition of the iron occurs because of the addition of uranium, there is a breaking up of the larger graphitic flakes, producing a closer grain and more homogeneous metal. Uranium iron is being used in glass-bottle molds and in the blanks and plungers in bottle machines. Uranium in iron seems also to increase considerably the latter's ability to take chill in casting.

Molybdenum.[25,26] There does not appear to be any reliable information concerning the effect of molybdenum on the combined carbon content. It would appear, however, that its general tendency is to reduce rather than increase the amount of carbide. Experiments carried out in this connection offer no evidence that small quantities of molybdenum affect the combined carbon content of the harder irons. It would appear, however, that on common gray iron molybdenum exerts an all-around improvement, giving the cleanness and fluidity of a soft iron with the properties of a cylinder iron. Molybdenum appears to decrease the chilling effect, although there is an increase in the hardness of the iron itself, and an improvement in the uniformity of hardness and solidity throughout the casting.

Chromium.[25,27] The effect of 0.11 per cent chromium on iron samples containing 1.30 and 2.0 per cent silicon and total carbon somewhat in excess of 3 per cent has been found to increase slightly the tensile strength, transverse strength, and Brinell hardness. With 0.78 per cent chromium, the tensile strength falls to that of ordinary gray iron and the transverse strength is little affected, but the Brinell hardness is raised 40 points. The value of chromium in cylinder metal is shown to be very limited in the manufacture of small castings. The chill is, however, affected. In ordinary cylinder iron its depth is increased from 0.068 to 0.90 in. by the introduction of 0.78 per cent chromium, a feature of great importance in the manufacture of chilled rolls, car wheels, gearwheels, and castings of any dimensions where toughness and hard wearing surfaces are of importance.

Acid-Resisting Irons. Truly speaking, these are not cast irons at all but hypoeutectoid steels as their maximum carbon content is only 0.80 per cent. They are, however, usually treated in the same light as cast iron, since because of their hardness they cannot be machined, but have to be used as castings only. These irons are primarily distinguished by their very high silicon content. It has been found that as far as resistance to action of acids is concerned, a silicon content of less than 12 per cent is ineffective. But the silicon is usually kept between 13 and 15 per cent. Metallurgically these alloys are solid solutions of iron-silicide and iron, the combination in which the carbon occurs being somewhat uncertain.[28]

Cast Iron as Affected by High Temperatures

Cast iron is affected by high temperatures in two ways. In the first place, it has been found that cast iron when repeatedly heated and the heating is followed by cooling, increases in volume. This phenomenon occurs even at temperatures below the lowest critical point, i.e., below 725 deg. cent. (1337 deg. fahr.), and causes the formation of fissures which produce porosity in the metal. Another effect of heat is that cast iron when at a high temperature is apt to lose considerable of its strength.

There have been a good many explanations of the causes of growth due to repeated heating and cooling. The most recent theories are (Okochi, Sato, and Kikuta) that the constitutional changes which occur at the critical points of heating and cooling set up considerable internal stresses, especially about the graphite flakes which are not subject to these changes. These stresses give rise to fissures and cavities in the neighborhood of the graphite flakes and produce an increase in the volume. This volume increase can be accelerated in an oxidizing atmosphere by the penetration of the oxygen into these fissures. The theory of these Japanese workers does not explain, however, the phenomena of growth below the transformation point, of which there is abundant evidence, such as, for example, the oxidation effect of steam at temperatures not exceeding 425 deg. cent. (797 deg. fahr.).

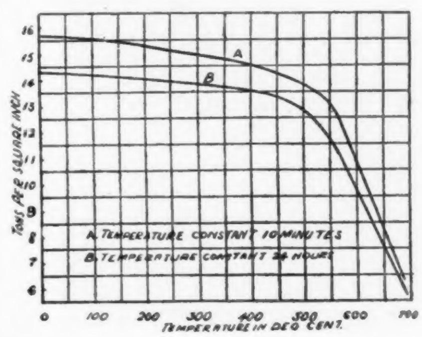

Fig. 19 Temperature-Strength Curve of Cast Iron (20 Per Cent Semi-Steel)

(Figure reproduced from p. 421 of *The Foundry Trade Journal* of Nov. 15, 1923.)

It has been found that within certain limits a higher silicon content is accompanied by greater growth. In fact, it is found that all the elements which, like silicon, precipitate graphite promote the increase of volume, while those elements which oppose the formation of graphite retard the process. Because of this, in cast iron where growth is objectionable, it is desirable to have a high manganese or chromium content (about 0.40 chromium), these elements forming stable carbides which resist changes. The formation of round graphite nodules instead of acute-angled flakes is also desirable.

Here it may be pointed out that cast irons low in total carbon have normally a very fine grain, and that on microscopic examination they show a highly divided graphite formed of very fine or curved lamellae, or of round nodules. This type of structure is frequent in semi-steel cast iron with high steel content (25 to 30 per cent), and particularly in cold and semi-cold blast metal. Moreover, it may be pointed out that this type of metal (total carbon low) may be obtained with a relatively high silicon content, two per cent or more. The result is, that with a suitable admixture of silicon, low total carbon can be obtained, while maintaining the combined carbon in the neighborhood of 0.6 per cent and even less.[29]

Strength at High Temperatures. Cast iron is superior to steel

for such purposes as Diesel-engine and general internal-combustion-engine cylinders, partly because of its better wearing qualities and also because of the greater facility with which complicated castings are made, which is due in part to the lower temperature at which cast iron has to be poured, its better ability to take the required shape, and also probably to the higher state of development reached by the cast-iron foundry as compared with the development of steel foundry.

The conditions of operation in internal-combustion engines, Diesel engines in particular, are such, however, as to submit cast-iron parts to considerable mechanical stresses at high temperatures. There have been a number of failures. An investigation by Arthur Marks[30] would seem to indicate that in bars which have been submitted to heating for 24 hr. at a constant temperature varying from 100 to 700 deg. cent. (212 to 1292 deg. fahr.) and then broken, the combined carbon at the center of the bar, where the fracture took place, showed no change. The breakage was therefore to be regarded as not associated with change of constitution, i.e., the production of weak graphite by the decomposition of iron carbides, but to the natural reduction of cohesion at high temperature. From this it would appear that this is not so much a problem for metallurgical research as a case for the engineers who will have to reduce the temperatures to which cast iron is subjected below the quite reasonable one of 550 deg. cent. (1022 deg. fahr.). From this investigation it appears the strength varied as shown in Fig. 19, namely, from 14.4 tons (of 2240 lb.) per sq. in. at normal temperature to 13.1 tons at 505 deg. cent. (941 deg. fahr.), but fell to 8.9 tons at 600 deg. cent. (1112 deg. fahr.) and 6.8 tons at 700 deg. cent. (1242 deg. fahr.).

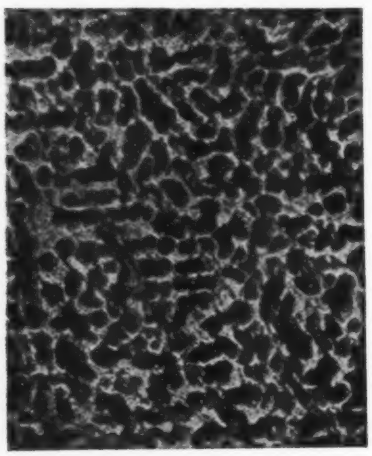

Fig. 20 Microstructure of White Cast Iron as Cast

(From Cast Iron in the Light of Recent Research, by W. H. Hatfield, Charles Griffin & Co., London, 1918.)

Another theory (André Levi) is that the increased volume following the first heat is due to decomposition of the cementite into iron and graphite, and the subsequent increase in volume is due to the combined effect of internal oxidation and the stresses produced in the metal at the transformation points.

Contraction of Cast Iron. Cast iron is one of the very few materials which expand on freezing, thus filling out the mold particularly well. Then contraction takes place. Recent tests[31] have shown that the initial expansion in both white and gray iron is in direct relation to the gas content of the molten metal. It seems that the sudden decrease of the ability of the iron to hold gas in solution which occurs at the instant of freezing of the iron is the cause of the expansion of the iron, the gas being thrown out of the solution and mechanically stretching the iron out.

Tests with white Swedish pig iron showed expansion decreasing with the degree to which the metal was degasified by suction. Pig iron rich in manganese, which, when melted in the air, solidified with very considerable expansion, did not exhibit any expansion, if, upon being melted, it was held for a short time under a vacuum.

As regards contraction, the difference between white and gray iron is primarily limited to the pre-pearlitic part, during which the secondary graphite is precipitated, while the post-pearlitic contraction is practically the same with gray and white iron. By regulating the rate of cooling, which means by controlling the temperature of the mold, it becomes possible to reduce the pre-pearlitic contraction to any desired extent or to eliminate it entirely. This makes it possible to eliminate the usual casting stresses and cracks in complicated shapes, and indicates another field where the application of metallurgical knowledge may bring important results.

Alloying elements may materially affect contraction phenomena by controlling the precipitation of the secondary graphite, it being generally found that the elements which assist the precipitation of the secondary graphite decrease the contraction, and those which hinder it increase contraction. This is why silicon, which promotes graphitization, decreases the contraction, while manganese produces an increased contraction through its opposition to graphitization.

Sulphur acts in a peculiar manner. In pure iron it decreases the contraction materially but in gray iron it promotes it rather strongly, because of its anti-graphitization action.

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Industrial Applications of Oil Engines

Their Employment as a Drive for Oil Pipe Lines, in Flour Mill Operation and Refrigerating Practice, and as a Drive for Pumps and Compressors—Summaries of Papers Presented During Oil and Gas Power Week at Kansas City

LAST month's MECHANICAL ENGINEERING contained summaries of a number of papers presented at various meetings throughout the country during Oil and Gas Power Week, April 20–25, and dealing with oil-engined locomotives, fuel oil and its supply, the trend of oil-engine design, oil engines in plants requiring steam, oil and gas conservation, and other related topics. Supplementing that publication, similar summaries of a group of interesting papers presented at one of the meetings of the Week, held in Kansas City, are given below.

Oil Engines as a Drive for Pipe Lines

F. Thilenius[1] presented a paper on this subject in which he said that the first successful pipe line was put into service in 1865, primitive methods of team and wagon and small-boat transportation being in use prior to that date, and the cost for hauling a barrel of oil half a dozen miles being often from three to five dollars. It now cost forty-seven cents to ship a barrel of oil by pipe line from Texas to Chicago. Pipe lines had grown rapidly in use, and the Prairie Pipe Line Company had now some 11,000 miles of line in operation with a capacity for delivering 200,000 bbl. per day. Present-day main-line practice, said Mr. Thilenius, consisted in laying single or parallel lines of 8-in. or 12-in. pipe, straight as far as possible, across the country, with a pumping station located about every forty miles and with receiving tanks at each station. At 700 lb. per sq. in. a single 8-in. pipe line carried 20,000 bbl. per day, with pumps of corresponding capacity. Roughly, 15 hp. was required for each 1000 bbl. per day of pump capacity. In early practice single-cylinder pumps were used, attached to a single-cylinder steam engine. Duplex pumps followed, driven by steam engines with practically no cut-off and a consumption of about 150 lb. of steam per hp-hr. Between 1890 and 1913 the more efficient triple-expansion steam engine prevailed, with a steam consumption of 15 lb. per hp-hr. After 1913 the oil engine came rapidly into use because of its superior economy, the steam engine requiring 1.5 lb. of oil per hydraulic hp-hr. while the oil engine consumed but 0.5 lb. Semi-Diesel engines were used and at first gave trouble owing to inexperience in operation, but by 1921 the transformation from steam to oil power was practically completed, with 28,000 hp. in Diesel engines in operation on the Southern Division of the Prairie Pipe Line Co. alone. Diesel engines, four-stroke cycle with compressors, were generally employed, with some thirty solid-injection engines. Operating difficulties mainly due to personnel had now been completely overcome and great reliability had been secured. Considering costs, Mr. Thilenius stated that the first cost of a Diesel engine was practically the same as that of a modern steam engine with all economizing accessories and of equal dependability. The oil engine needed a little more attention and frequent adjustment, but this could be taken care of by a spare unit, which, however, must be included in first cost. The labor required compared on the whole favorably with that necessary for a steam engine. Cylinder lubricating oil had to be of a higher quality for the Diesel. Maintenance was on the whole higher, and this must be balanced against lower fuel consumption. While no very exact figures on maintenance were available, recent improvements in design had considerably reduced such costs. Also Diesel engines were no longer used up to their rated capacity, for under full-rating use exceptionally favorable operating conditions were required. Among the improvements in design which had lowered maintenance charges were: improved lubricating systems, with filters remodeled so as really to settle and filter the oil while hot and remove the water, while centrifugal separators had been installed at the larger stations to help out the filters; treatment of the cooling water before use and periodic removing of the mud and scale had reduced the number of broken cylinder heads, cracked liners, etc.; concrete reservoirs had replaced earthen water reservoirs; the use of pyrometers had enabled the operator to balance the load better between cylinders; air-washing devices and fuel-oil cleaners would probably come into use; there had also been improvements in design of the engine parts themselves. Another economic factor in favor of the Diesel engine in addition to its greater thermal economy was the possibility of utilizing the heat of the exhaust. Tests had shown that 43 per cent of this heat could be readily reclaimed. While no attempts had been made to secure power from the heat of the exhaust, useful heating possibilities had been established.

J. T. English,[1] discussed the pumping stations on the northern division of the Prairie Pipe Line Company which had 3000 miles of pipe lines and 18 pumping stations. In that division as in the southern division the year 1913 marked the first installation of Diesel engines. Now the various pumping stations were equipped with Diesel oil engines and vertical triplex power pumps, or with oil-fired tubular boilers. At one station coal screenings were used for fuel under water-tube boilers. Of the aggregate of 38,000 hp. 23,000 was oil-engine equipment and 15,000 steam-plant equipment. Mr. English gave some interesting figures with regard to the operation of the oil engines. The fuel consumption had been very satisfactory. Of total operation costs, only 16 per cent had been fuel cost; 50 per cent had been the figure for fixed charges, 22 per cent for regular station labor, and 5 per cent for engine repairs. As to engine repairs, 20 per cent of the repair costs had been due to valve trouble, 12 per cent to pistons, 10 per cent to cylinder heads, and the remainder divided over the engine. The percentages referring to fixed charges and repair costs were now subject to downward revision due to the comparatively low first cost of oil engines, and to the many later improvements in oil-engine design. Repair costs had been largely reduced by using a load factor of not more than 85 per cent. Mr. English cited, however, a case where the low price of coal screenings had made this type of fuel cheaper than oil in oil-engine equipment, and concluded his paper by saying that the economic superiority of the oil engine was not assured under all conditions.

Oil Engines in Flour-Mill Operation

Chas. Dalrymple[2] read a paper containing much first-hand information in regard to experiences with flour-mill power plants. Water power, he said, was not always available where wheat was plentiful, and failed during high water or drought. For this reason flour mills had been quick to take up the Corliss steam engine, which had been most satisfactory during the days of cheap fuel. During the last twenty years flour mills had developed from capacities of two or three hundred barrels per day to units of several thousand barrels' capacity, and the cost of fuel had doubled. Accordingly of recent years very few Corliss engines had been installed, and power had been furnished either by Diesel engines or by electric power purchased from power stations. With electric power small and unexpected disturbances were a source of much annoyance to millers, because a flour mill might indeed contain hundreds of different machines, yet the plant must operate as a unit. Every machine was connected through spouting and if the plant as a whole slowed down or stopped for only a moment, it was necessary to shut down entirely until the machines and spouting could be cleared of choke-ups. Some of the electric line disturbances could have been avoided if the transmission lines had been equipped with automatic switches to isolate the trouble, but electric companies had generally not thought this service to be worth while. After a complete investigation in 1922 the Red Star Milling Co. decided to install Diesel engines, the slightly

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[2] Chief Engineer, Red Star Milling Co., Wichita, Kan.

higher investment as compared with the steam turbine being offset by lower fuel consumption. Two 750-hp. McIntosh & Seymour engines equipped with 750-kva. generators furnishing power for two mill units of 3000 bbl. daily capacity were installed. Since January, 1924, the performance for ten months had been entirely satisfactory. The speed had been so uniform that choke-ups in the mills had been practically eliminated, although the plant had been operated at full load with no spare engine. Success in operation was largely attributed to the use of specially trained Diesel-engine men. Fuel oils of various densities and complexions had been used, and even in hot weather it had been necessary sometimes to apply heat to the oil to make it thin enough for pumping. The conclusion was that anything that looked like oil and could be pumped when heated, could be classed as Diesel-engine fuel. It had been found that even with an excellent fuel oil particularly free from any sediment or foreign matter, the fuel pump would accumulate a deposit and there would be a tendency to lift the valves from their seats unless the percentage of salts was below a certain point. Water in suspension in the fuel gave trouble if no steps were taken to settle it out. Fuel containing carbon residue—to a large percentage due to modern methods of cracking—could be used provided the strainers were cleaned oftener and the fuel-spray nozzles given attention at shorter intervals. The complete records for 1924 showed a total output of 4,354,295 kw-hr. with a consumption of 402,917 gal. of fuel, giving approximately 11 kw-hr. per gal. The total operating costs were 6.16 mills per kw-hr. This left a margin below the cost of purchased power for interest, depreciation, and taxes more than sufficient to justify the investment.

The Diesel Engine as a Drive for Pumps and Compressors

F. I. Kemp[1] read a paper on this subject in which he first took up the problem of pumping water and compared the oil-engine drive with electric power. The comparison with electric power was justified from the trend of current practice and the fact that the smallest Diesel engine was more economical than the most highly developed steam equipment, and the largest Diesel engines could reach a maximum of 230,000,000 ft-lb. per million B.t.u. or about $^1/_3$ better than the best steam pumping unit ever built. Several types of pumps could be selected for connection to the engine, these being centrifugal pumps, vertical triplex power pumps, and horizontal duplex or triplex power pumps, with a variety of drives and gearings to the engine possible in each case. Mr. Kemp took a practical example of a city where 700 gal. per min. were required to be pumped against a total head of 307 ft., with 16 hours per day of operation. Selecting a vertical single-acting triplex power pump connected to an oil engine through a friction clutch and a single spur-gear reduction, and assuming a pump efficiency of 83.5 per cent, Mr. Kemp figured that a 75-hp. engine was necessary, operating a 12 by 12-in. pump at 41 r.p.m. The first cost of the equipment including foundations, buildings, oil-storage tanks, and complete erection, would be $16,000. The total yearly cost based on oil at 6 cents per gallon, labor at $300 per month, and including fixed charges, would be $7792 per year. The total pumpage per year based on 16 hours' operation per day would be 245,000,000 gal., giving a total cost of 3.02 cents per 1000 gal. pumped. This Mr. Kemp compared with an electric-motor-driven triplex pump for the same conditions. The first cost he estimated at $85,000 completely erected, and the total yearly cost exclusive of power cost at $4800, giving $2600 per year available for current on the same basis as with the oil engine. The current consumption per year would be 321,000 kw-hr., and the electric cost would have to be 0.811 cent per kw-hr. if the oil engine was to have any serious competition. When the oil-engine installation was compared with an electric-motor-driven centrifugal pump with certainly the lowest first cost of any type of installation and the possibility of reducing labor charges to a minimum, it was found that electric current would have to be purchased at 1.69 cents per kw-hr. From this Mr. Kemp concluded that the oil-engine-driven pump was almost certain to be cheaper than the electrically driven pump. Remote control for a water-pumping plant also meant increased insurance rates for a municipality. Similar analyses which he presented for much larger water-pumping installations led to the same conclusion, namely, that the Diesel-engine installation was more economical. Mr. Kemp then considered the problems of operating air compressors in a similar fashion. Taking the case of a central air plant supplying air to a group of mines in the lead and zinc district where it had been found necessary to use three compressors each delivering approximately 2000 cu. ft. per min., it was found that the necessary compressor size was a $26^1/_2$ by 18-in. duplex two-stage machine arranged for direct connection to a 400-hp. oil engine. With an oil engine the cost of the installation would be approximately $145,000 and the total yearly cost based on 5-cent fuel, 9 hours per day, and 300 days per year, $32,340. With full load assumed for this period the cost per 1000 cu. ft. of free air actually delivered was found to be 3.12 cents. A synchronous-electric-motor-driven installation having a first cost of $60,000 would have to be supplied with electric power at 0.683 cent per kw-hr. to give an equivalent cost. Records of such an installation confirmed the view that the oil engine was as superior to electric power in driving air compressors as it was in driving pumping machinery. Mr. Kemp concurred with other authors presenting papers during Oil and Gas Power Week in emphasizing the need of skilled operators for Diesel engines.

[1] Manager, Kansas City Office, Worthington Pump and Machinery Corporation.

The Oil Engine in Refrigerating Practice

Geo. M. Kleucker,[1] in a paper on this subject, said that the adoption of oil engines in refrigerating and ice-making plants dated back only about twenty years. Steam engines were at first employed exclusively because the exhaust furnished the necessary condensate for the distilled water. The Corliss-valve-gear type of steam engine as directly connected to the ammonia compressor was considered economical enough as the exhaust steam was much less than that required for the ice-making capacity of the compressor to which it was connected, the remainder of the condensate being made up from various auxiliary engines. Steam economy was not considered as important as the production of clear distilled-water ice without a core, in blocks of 300 to 400 lb., the ice being made in freezing cans submerged in brine. When it was discovered that clear "raw-water" ice could be made in plates of two to four tons, more economical steam equipment with high-pressure boilers and compound condensing engines came into being. The "plate plants" had their real beginning in 1917, during the war, when shortage of coal furnished the impetus for the change from distilled-water to raw-water plants. At the same time, shortage of coals likewise led ice manufacturers to make contracts with electric power companies. The exact metering of the power led in turn to an era of greater economy. The passing of the individual steam plant also resulted in the advent of the oil engine. At first, owing to lack of experience, many difficulties were experienced with oil engines, particularly as ice-making machinery was not purchased together with the power equipment. At the present time, the oil engine was working its way to the front by virtue of lower costs and freedom from shutdowns entailed by the use of electric power. While maintenance costs were excessive with some oil engines, the impression that it was always excessive is erroneous. A comparative analysis made by Mr. Kleucker for an ice plant of 30-ton capacity using fuel oil at 5 cents a gallon showed a saving of $4400 per year as compared with an electrically driven plant, the electric power costing 1.75 cents per kw-hr. Discussing the mechanical features of oil-engine equipment, Mr. Kleucker pointed out that the oil engine being on the same shaft as the compressor, very compact units were possible. The oil engine equipped with variable-speed governors afforded a flexibility comparing very favorably with steam-operated compressors. The fuel consumption per ton of ice was about four gallons, giving a fuel cost of about 12 cents per ton of ice. Labor-saving devices were being given much attention. Freezing cans were now being hoisted out of tanks in multiples, and ice-scoring machines sawed the blocks into predetermined weights which insured the customer of an honest weight.

Howard P. Morris[2] read a brief paper on specifications for Diesel oil engines. He thought that specifications calling for prices on steam equipment as well as Diesel equipment could be eliminated by

[1] De La Vergne-Machine Co., New York, N. Y.

[2] Representative of Fulton Iron Works Co., St. Louis, Mo.

the preliminary study of fuel costs to determine the more economical type of equipment. He pointed out some objections to engine bids through general contractors, who were not always competent to erect and had to include handling charges in their bid. He did not think it advisable to dictate to builders of Diesel engines as to design and material; formal, unrestricted bids by the engine builders might contain much useful information. After a discussion of bidder's checks, time of day most suitable for bidding, etc. Mr. Morris made some suggestions for reducing costs. The purchaser when contemplating a Diesel-engine installation should pour his own foundations and purchase switchboards and other accessories direct so as to avoid the engine builder's handling charges. Similar procedure might be followed with cooling towers, fuel tanks, piping, etc. It was always useful, however, to consult the engine builders regarding the cooling system. When the water was very hard a closed system should be installed, or if a closed system was not desired, a water softener might be placed in the plant to supply the make-up water.

The World Economic Situation

By G. A. O'REILLY,[1] NEW YORK, N. Y.

THE extreme importance of a careful, serious, and continuing study of world economic conditions—and this naturally includes world politics—is one of the unmistakable things which has come with the development of recent years, and particularly has it come to our own country. Of course, it has always been desirable that nations should keep in touch with each other and be familiar with the great currents which run constantly between them, but the war and the things which have come since have given a new importance to this, have made it imperative. If some nation wishes to try a spell of world isolation, that is the affair of that nation. Of course, it will not work, but the experience may be helpful as an object lesson to other nations. But even if we could assume that self-isolation would work, we could not well imagine it upon any basis other than one which would include a rather complete knowledge of this world from which the particular nation trying the experiment was to be isolated. Whether a nation's position is to be offensive or defensive, or coöperative in the fullest sense, the need for a knowledge of the other fellow's situation exists.

I wonder if it would be too much to say that if nations had been properly informed concerning each other, the war might have been averted. It has become so popular to assume that the war was inevitable that it is easy to overlook important elements which existed but which were not of general knowledge; elements which, upon the theory of a broader dissemination of world economic information, might have made the history of the past ten years very different from what it has been.

CONDITIONS BEFORE THE WAR

In Russia, before the war, there existed a condition, part economic, part political, part sociological, but, considered in terms of world interest, altogether economic, which was not understood either by the Russian people themselves, who as usual were not interested, or by the then Russian Government, which had expected nothing more serious than a possible palace revolution, or by the people or governments of the more highly organized countries of the world, who had come to accept Russia as a fact which somehow or other would muddle along without serious disturbance to the world. Of course, all were wrong—no constructive world effort was made—and as a result we have a Russia in chaos with a more or less serious effect upon the economic condition of every country under the sun.

Before the war the term "national bankruptcy" carried with it a peculiarly terrible significance, and expressed a fate from which all good nations were to enjoy immunity. Today most of these nations, their governments at least, are bankrupt. And the strangest part of it all is that there are excellent reasons for believing that, in spite of their practical bankruptcy, they will come through their difficulties, every one of them. But in the meantime the economic condition of the world will have been dragged over a rough and rocky course.

The world since the war is filled with illustrations of "what might have been" had we only known in time. Each tells, in a measure, the story of a possibly unconscious national effort toward self-isolation. In finance, in commerce, the activities of the particular nation may have been world-wide, but it did its studying at home. Striking occurrences abroad were able to attract attention. National calamities which made an appeal to the humanity of nations rarely failed to bring forth a response. Conspicuous manifestations of good-will or ill-will between nations received at least their proper share of attention. But the great currents of world movement which flow constantly everywhere and which are a part of the economic life of all nations flowed on, and were noticed only when, coming to the surface in one country or another, they disturbed the ordinary course of things sufficiently to demand attention.

This need not mean that the world is all wrong or that a great discovery has been made or that national habits are to be changed. It is not my purpose to underestimate national intelligence anywhere. I only hope to be able to assist in the interpretation of the events of the past few years and to apply, principally to our own national situation, the object lessons to be found in these events.

THE AWAKENED AMERICAN INTEREST IN WORLD ECONOMIC STUDY

The United States, probably the least understood and appreciated of the nations, seems peculiarly in need of the development of a more serious attitude toward this study of world conditions. It is not enough that we think long and hard. We also must dig for information, and in a much broader field than that to which we have confined our efforts. With us there no longer is any possibility of isolation. Our place is in the big picture, the world picture. The other nations realize this and have realized it for years. It is about time that we began to realize it and build upon it. We are in this picture financially. We are in it commercially. We are in it scientifically. And we are in it to stay. And all of this entirely apart from any question of participation or entanglement in the political squabbles which the world, and particularly Europe, apparently insists shall occupy the center of the stage.

And the beginnings of a newly awakened American interest in world economic study are evident. Our interest is not academic but a fairly definite detail in a constantly improving scheme of American business economics. We are beginning to see our world more clearly than formerly, and also our United States. A few years ago Europe and the Far East were places far away and, save to those who had definite dealings with them, vague and meaningless—places to visit when we could, places filled with interesting peoples and institutions and traditions, but not in any essential way related to the active, every-day life of our people. We traveled abroad for recreation, for amusement, for culture, and boasted that we never mixed business with pleasure. Now the letters of credit we carry on our pleasure trips are apt to be large enough to include also business necessities, which have a way of intruding regardless of the nature of the trip.

Formerly, to the average American—and this included the business man as well as the others—Europe meant Paris, the Riviera, Switzerland, Venice, Berlin, Monte Carlo. These places still are popular, but so are an almost infinite number of other places which have no particular significance beyond their bearing upon

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Address delivered at a meeting of the Hartford, Meriden, and New Britain Sections of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS at the City Club, Hartford, Conn., May 7, 1925.

things economic. Then the American traveler visited cathedrals, museums, theaters, summer and winter resorts; now he visits these possibly, but also and more particularly banks, government offices, industrial establishments, commercial concerns. Then, in his travels, he drifted, or at least did not employ definiteness of movement beyond possibly "following the man from Cook's." Now it is nearly sure to be a matter of arranged schedules, business appointments, and the other evidences of seriousness which we associate with the idea of business.

Our study of world economics has not yet proceeded far enough to have become established as a national habit. We still seem disposed to interest ourselves only in the more spectacular features of the case—conspicuous movements, world economic disturbances sufficiently serious to suggest world danger, or danger to American interests particularly. But the common course of things has not yet succeeded in asserting its full importance.

And still, it is this common course of things in the world which deserves the greatest share of our attention. The conspicuous occurrence, and particularly if it suggests danger, sounds its own alarm and the world takes heed and provides against it. The selection of a labor government in England needed no study on our part or on the part of any nation. It told its own story and provided its own answer, and, fortunately for the world, that answer spelled safety and conservatism, and the checking of one of the world's dangers. Russia, too, in its chaos and apparent hopelessness, has provided, if not the answer, at least a fairly reasonable explanation of its difficulty. The Mussolini movement in Italy was easy to understand, required no particular study once it was started, and the problem, because it is so conspicuous and its danger so fully realized, now is Italy's, not the world's.

The Importance of Studying the Inconspicuous Economic Movements

But the inconspicuous, from day to day, movements throughout the world which sound no alarm, give no warning, and unless we seek it, give no suggestion as to their true meaning or ultimate effect, these are the things which are responsible for most of the success and most of the failures among the world's nations, depending usually upon the seriousness with which they are studied and analyzed. The course of the franc in Paris, the resumption of the gold standard in London, the establishment of an English capital naval base at Singapore, Japanese immigration into the United States, the political situation in Bulgaria, the recognition of Soviet Russia by leading nations, the relation of European countries to each other in the troubled area of North Africa, the development of scientific research in Germany, the practical difference between the bankruptcy of a people and of a government—these are some of the questions which well may form the basis of profitable world study in the interest of our own economic situation.

It is unfortunate that, in our study of world conditions and of conditions in our own country, we devote so much time to the search for points of weakness and so little to the search for points of strength. True, we should not delude ourselves as to the seriousness of weaknesses in our situation or of the difficulties which may come; but neither should we delude ourselves as to the fact of our power. And the point of power is so much easier to overlook than the point of weakness. The one we take for granted and forget; the other, in a sense, sounds its own alarm and demands attention. Then, too, we have formed the habit of considering confessions of weakness as meritorious and references to power as boasting, a most unfortunate habit.

The Realization of America's Power

Because of present world conditions and our own national position in the world, there could hardly be a more important subject for consideration than power, our own power. The facts of the world case should be realized and build upon. The war brought into the world many new and surprising things, and among the most striking of these was the discovery of the unsuspected power of the United States. The world found, and with distinct surprice, that we are a really great nation—in point of power, the greatest. The world had not always seen us in this part, nor had we ourselves. Big and strong and healthy we were, they had said, but crude and awkward and unskilled in many of the things that go to make up the total national greatness. They were frank and undoubtedly honest in this opinion, which included our army and navy, our banking system, our methods in trade, our knowledge of fundamental economic, science, art, and letters. They conceded to us some of the advantages of youth and all of its disadvantages.

But this has changed. Our case has been made clear, our position among the nations has been established. Our part in the war, and in a hardly less difficult world since, has shown conclusively our true place in the world picture. The last vestige of world doubt was removed in the effort which found final expression in a Dawes Plan and an actual start toward European rehabilitation. We must not delude ourselves with the belief that it is a case of the world falling in love with us—such things do not happen. But the nations have come to realize the power of our country and will respect it just so long as we show them the way. Nevertheless the world has a way of forgetting even so serious a thing as national power, and we must realize that America's place will be maintained only if we maintain it.

The World Looking to the United States for Economic Leadership

It does not seem too much to say that just now the world looks to the United States for leadership, economic leadership, which, if we are wise, is the only leadership we shall attempt. It is fortunate for us and possibly for the world that our new position involves no suggestion of political leadership. Our capacity in this direction will be sufficiently taxed by the problems at home. And the fact that we are not in world politics makes our world economic position all the stronger. The money the world borrows from us—and it has borrowed much and will borrow more—is excellent money, absolutely untainted by any ambition in the direction of world domination, military aggression, or acquisition of territory. And the credit we have been extending to the world so freely is—just business. There is nothing of the political about it. Is it strange that, in an economic sense at least, we are exceedingly popular just now?

But we should understand that our national power must be safeguarded if we are to keep our place in the economic procession. In a world so filled with competition, the order of things has a way of changing. It will not be enough that we sit around at home and dream about this power that is ours. We must go out and study the world's ways. Economic currents must be traced wherever they are and in whatever direction they run. And they are not always easy to find or analyze. World tendencies must be considered in terms of their bearing upon our national interests. Upon our success in doing this will depend our own national economic safety and much of America's possibility of helpfulness in the world.

Tide-Power Plant for Maine

A TIDE-POWER project for the development of 500,000 to 800,000 hp. is proposed by harnessing the tides of Passamaquoddy Bay. The state legislature of Maine and New Brunswick authorities have approved a charter for the project and preliminary work is being done now, but it is expected that 5000 men will be employed for five years for the completion of the work. The waters will be held in check by immense dams so constructed that the power station will utilize the water in both the ebb and flood tides. The rise of the tide is twenty-two feet. The charter contains a provision for ratification or rejection by popular vote, and at a special election which is to be held in September the vote is to be cast.

Surplus power, not required in Maine, is to be transmitted elsewhere. The project was at first opposed by local power companies, but they have been won over by the argument that the proposed plant will manufacture power cheaper than they can so that they will be able to buy whatever power they need. Roadways are to be built on top of the dams for the use of the public in Maine and New Brunswick. The development is proposed by Dexter P. Cooper, an engineer, of New York, and the construction cost is expected to amount to $100,000,000.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

The Present State of Knowledge as to Heat Transfer in Gases

IN THE January, 1925, issue of MECHANICAL ENGINEERING (p. 41) there appeared a paper by Alfred Schack entitled, New Data on Heat Radiation, based on investigations carried out in Düsseldorf at the Heat Research Bureau of the Association of German Iron Makers. In that paper reference was made to the then unpublished investigations of Lent and Thomas, who had carried out their work on a blast-furnace flue at the Rhine Steel Works.

The present paper is written by the first of these two investigators and, in the words of the author, "attempts to present in a comprehensive manner the present state of our knowledge on heat transfer without frightening the reader away by an excessive use of mathematical formulas and big words."

The great difficulty of all investigations of heat transfer in gases lies in the fact that, contrary to what happens in the investigation of heat transfer in liquids, one is constrained to deal simultaneously with three phenomena—conduction, convection, and radiation—so closely intertwined that it becomes impossible, or at least inordinately difficult, to separate them.

According to an earlier investigation by Nusselt, the heat transfer through convection above a certain velocity, which he called "critical," is a function of velocity of flow of the gas, pipe diameter, coefficient of heat conduction, and temperature varied in such a manner that the coefficient of heat transfer increases with the 0.8 power of the velocity. The validity of Nusselt's formulas for engineering purposes has been questioned for a long time. Recent partly unpublished investigations would seem to indicate that the variation of heat transfer first begins at velocities higher than those which have been considered "critical" hitherto in technical literature, so that, for example, in the case of heat transfer from firebrick to air and from flue gas having a temperature of about 500 deg. cent. (932 deg. fahr.) to brick in regenerators and recuperators, the coefficient of heat transfer at ordinary velocities may be considered as invariable; while at higher velocities such as are encountered, for example, in the tubes of waste-gas boilers used in connection with gas engines, the coefficient of heat transfer increases with the 0.8 power of the velocity.

The Nusselt formulas are strictly valid for flue gas up to temperatures of 500 deg. cent. (932 deg. fahr.). Above this limit, however, the influence of heat transfer by radiation from gases becomes noticeable and rapidly grows to several times that of heat transfer by convection. The best-known process of heat exchange is that through radiation between solid bodies, for example, between two opposed parallel plates separated by a stationary layer of dry air. Dry air is practically completely transparent to all rays. The conditions obtaining in radiation depend in such a case entirely on the state in which the solid materials happen to be, and there is no external impetus with the rise of temperature to emit energy in the form of waves. In such a case the equalization of temperature proceeds as a function involving the fourth degree of the absolute temperature (Stefan-Boltzmann law).

The ability of a body to emit rays bears a certain functional relation to its ability to absorb rays (compare Kirchhoff's law). The magnitude of the energy radiated outward is here dependent on the character of the surface, which, under certain conditions, may reach the maximum value. (The coefficient of radiation of a perfect black body is equal to 4.9 cal. per sq. meter per hr. per deg. cent.) Usually it has only a fraction of this value. Measurements of the coefficients of radiation, especially at high temperature, are difficult to make and scarce, and the correctness of some of the more modern data is in some doubt. The strength of the light effect on the eye is not always a good measure of the radiated energy. It is, however, possible to determine this quantity by means of proper measuring instruments, such as optical pyrometers, and by this method to estimate the surface temperature governing the radiation. Total-radiation pyrometers rarely permit determining the right temperature, because of the fact that the blackness of the body on which the measurements are being made varies and is difficult to establish.

If, now, in the case of the two parallel plates referred to above, the layer of air be replaced by a layer of gas which itself has the ability to emit and absorb radiations, it will be found that the gas layer is heated by the radiation from one plate, and that only that part of the radiation from the first plate is likely to reach the second plate which the gas has not absorbed in the interim. Gases, however, can absorb only rays of certain predetermined lengths. Among those which are capable of emitting heat rays and absorbing them to the same extent, physicists have for a long time known carbon dioxide, water vapor, hydrocarbons, and possibly carbon monoxide. Dr. Schack was the first to prove mathematically that in the temperature range above 500 deg. cent. (932 deg. fahr.) these radiation phenomena in gases may be of supreme influence on the method of occurrence of heat transfer. In this way he explained the hitherto incomprehensible high coefficient of heat transfer found in boilers and furnaces. (Compare Schack's paper in MECHANICAL ENGINEERING, Jan., 1925, pp. 41–44.)

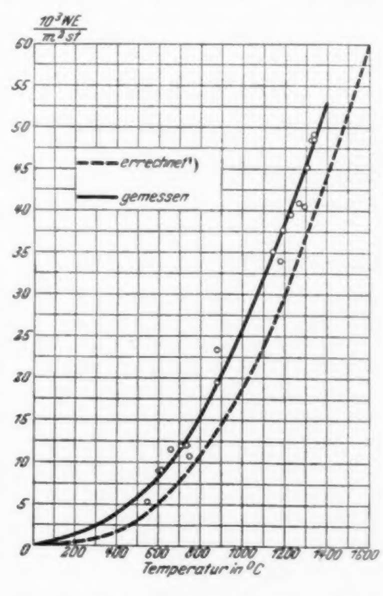


FIG. 1 RADIATION IN A LAYER OF GAS 1.23 M. (48.42 IN.) THICK CONTAINING 21 PER CENT CO_2 AND 2.4 PER CENT WATER VAPOR

(Broken line, calculated values; solid line, measured values; W. E., Calories; m^2 = square meter; st. = per hour.)

Without going into the whole theory of gas radiation, the author points out that the gases occurring in firing, including nitrogen and oxygen, are all capable of emitting and absorbing radiations and that temperature, thickness of the layer, and concentration of the gases capable of radiation have an influence on the duration of the radiation exchange. Furthermore, in order to correct an erroneous but often expressed opinion, it may be well to point out that the exchange of radiations between these gases and solid bodies does not follow the Stefan-Boltzman law, but proceeds in accordance with a complicated exponential function derived by Schack. In order to give an idea how radiation works out in a gas layer, the broken line of Fig. 1 shows the calculated magnitudes of radiation from a gas layer 1.23 meters (48.42 in.) thick and containing 21 per cent carbon dioxide and 2.4 per cent water vapor. The solid-line curve in the same figure gives data of radiation as determined by actual measurement of the same layer of gas, and the fairly close agreement between the two is worthy of notice. The values determined by measurement are on an average 10 per cent higher than the calculated values, which is reasonable, as Schack used minimum values for his calculations. The as yet unpublished investigations of the research laboratories of the Siemens-Schuckert Works as well as the investigation of E. Goebel at the Julien Iron Works and of Dr.

W. Heiligenstaedt at Rheinhausen, indicate that the Schack formula gives a correct foundation for the computation of radiation from gas flames, carbon dioxide, water vapor, heavy hydrocarbons, and carbon monoxide.

The measurements of heat radiation in an open-hearth furnace has, however, given higher values than those obtained from the Schack formula. This may be explained, however, by the presence in the open-hearth flame of radiations from vapors produced by metals and slag, as well as by the influence of the very finely divided particles of carbon produced by the decomposition of tar and hydrocarbons.

It has been stated elsewhere that with increasing subdivision of the heat-absorbing body, the heat transfer through convection increases. This leads one to the conclusion that the finely subdivided carbon as long as it does not burn up is itself a carrier of heat. It absorbs the heat by convection and emits it by radiation. This would confirm the view of metallurgists to the effect that a so-called "luminous" flame is more efficient in the way of heat transfer than a nonluminous one, although actually the luminosity of the flame as such has often nothing to do with the power of radiation and when taken by and in itself is apt to lead to erroneous conclusions. (Dr. of Engrg., H. Lent, Dinsburg-Ruhrort, in *Stahl und Eisen*, vol. 45, no. 24, June 11, 1925, pp. 938–940, 1 fig., *tA*)

Short Abstracts of the Month

AIR MACHINERY (See Power-Plant Engineering: New Union Station in Chicago)

ELECTRICAL MACHINERY

The Hunt Self-Paralleling Alternator

An ordinary polyphase induction motor if driven at a speed exceeding that of synchronizing will act as a generator, returning energy to the line from which it would be supplied when running as a motor. An induction generator requires no synchronizing and the frequency of the output is always equal to that of the line, independently of the speed of rotation so long as this exceeds the synchronous state. The drawback of such machines is that they are not self-exciting, which is a serious matter as the excitation current is comparatively large, amounting to something of the order of 30 per cent of the full-load output of the machine as a generator. Such a generator can therefore only be used in parallel with synchronous generators to supply the magnetizing current.

The alternator recently developed by L. J. Hunt (4, Broad-Street Place, E. C. 2, London), is so arranged as to be a combination of an induction and synchronous generator in a single machine so that it does not need to be synchronized before being connected in parallel with other machines and does not need external supply of excitation current.

Although in the actual machine the synchronous and asynchronous portions are combined in a single stator and rotor, the principle on which it operates can best be understood by considering the two parts separately, as drawn diagrammatically in Fig. 1.

As there shown, the excitation current is supplied by a continuous-current generator to an 8-pole stator winding, generating current at a frequency of $16^2/_3$ (in the original, 16) cycles per second in a rotor driven at 250 r.p.m. This current, supplied to a 16-pole rotor winding of a separate machine, would produce a field rotating at 125 r.p.m., even if the rotor itself were stationary. If, however, the rotors of the two machines are mechanically coupled so that both rotate at the same speed, i.e., 250 r.p.m., a field rotating in space at 375 r.p.m. can be produced by the rotor of the second machine. Under these conditions, if the stator of the latter is provided with a 16-pole winding, alternating currents with a frequency of 50 cycles per second will be generated in that winding, and these currents can be tapped off to supply the line.

In practice, instead of employing separate mechanically coupled machines, the generator has a single stator core provided with windings giving the resultant of the 16-pole and 8-pole fields. The construction adopted is similar to that of an induction motor, the stator and rotor cores being built up of laminations provided with slots for the windings, the slots of the stator being open and those of the rotor almost closed. In the rotor slots is a simple short-circuited winding producing the same effect as two independent 16-pole and 8-pole windings.

Several alternators of this type of considerable size have been built and installed in various places in Great Britain, and some tests were carried out to demonstrate the self-paralleling properties of these alternators. The installation where they were made consists of two machines of 160 kw. capacity at a power factor of 0.8, each machine being directly coupled to a four-cylinder horizontal gas engine running at 250 r.p.m. In the test the two sets were at first running normally; next, first one cylinder and then two cylinders of one of the sets were cut out and the two machines remained perfectly in step, though one engine fired on four cylinders and the other on two. Subsequently the sets were run with three cylinders in each set and still kept perfectly in step. This means that by employing self-paralleling alternators the necessity for applying a very uniform driving torque is avoided and the cost of constructing internal-combustion engines to drive them can accordingly be materially reduced.

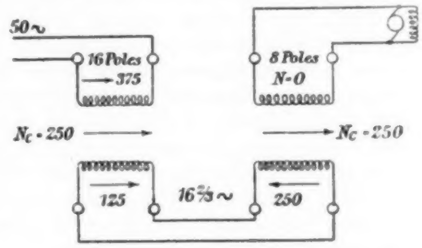


Fig. 1 Diagram Illustrating Principle of Operation of the Hunt Self-Paralleling Alternator

In another plant one of these alternators having a capacity of 210 kva. is running satisfactorily in parallel with an ordinary synchronous alternator of the revolving-field type. Hunt alternators have also been employed in hydroelectric power plants and an interesting installation of this kind, at a mill in Scotland, may be briefly referred to. In this case, during the winter months there is plenty of water available and the machine, driven by a Boving hydraulic turbine, runs as a generator in parallel with the supply company's mains, power being delivered to the town when the mill is shut down. During the summer there is insufficient water to drive the turbine and the alternator is then run from the mains as a synchronous motor, driving the mills and also improving the power factor. At other times the supply of water varies rapidly, and the machine changes equally rapidly from an alternator to a synchronous motor, and vice versa, without any signs of instability, the change being noticeable only by the reversal of the energy meter. The most obvious application for the new type of alternator, however, is in connection with internal-combustion engines, since these machines can be driven by single-cylinder or two-cylinder engines, and operated in parallel without any of the difficulties experienced in running synchronous alternators in this way. (*Engineering*, vol. 119, no. 3102, June 12, 1925, pp. 723–725, 4 figs., *d*)

ENGINEERING MATERIALS

Alloys of Diamond Hardness

A material which is claimed to be able to replace the diamond in core drilling and stone cutting has been developed in Germany. It is supposed to consist of a mixture of tungsten carbide and tungsten, and is called "phoran." It has a melting point of 3000 deg. cent. (5432 deg. fahr.), does not soften or fuse at any lower temperature, and therefore cannot be forged. It possesses a hardness on the Moh scale between 9.8 and 9.9, the diamond being 10, and is of considerable density. (*The Iron Age*, vol. 116, no. 3, July 16, 1925, p. 151, *d*)

Solomite: A Building Material Made of Compressed Straw

Description of a material invented by M. Tchayeff and now being exhibited as part of a number of buildings at the Exposition of Decorative Arts in Paris. In this process straw is utilized not in the form of blocks as previously but in the form of boards about

5 cm. (2 in.) thick. In these boards the separate straws are laid parallel to each other, strongly compressed, reinforced by steel hoops 2 mm. (0.078 in.) thick, and transversely tied by iron wire. The usual way in which this material is made is in rectangular boards 2.8 m. by 1.5 m. (9.18 ft. by 4.92 ft.). Because of the compression and reinforcement the boards possess great cohesion and appreciable resistance to bending.

When used in buildings the material must be protected by an outside layer of mortar, laid on in several layers. The use of a cement gun is particularly suitable for this purpose.

The new material is said to be a good noise and heat insulator, and if properly used does not rot as one might be afraid it would. Also straw is not attacked by various parasites.

One particular application of this material is to fill in the space in buildings where double walls are used and where the air layer contained between the walls is not sufficient to provide insulation against the loss of heat. This is particularly suitable for the new type of steel houses in which the external walls are formed of steel sheets and the interior walls of asbestos board. (*Le Génie Civil*, vol. 86, no. 24, June 13, 1925, pp. 588–589, *d*)

Corrosion of Non-Rusting Iron and Steel Alloys at Different Temperatures

The resistance of different varieties of steel to attack by chemical reagents was tested at the ordinary temperature, at 60 deg.–80 deg., and at 400 deg. cent. Highly alloyed quaternary steels possess the highest resistance to corrosion. Chromium steel containing at least 11 per cent chromium and not more than 0.2 per cent carbon is also highly resistant, and the resistance is increased by increasing the chromium content or by the addition of nickel. Nickel steels are attacked more readily than chromium steels by spring water, sulphur dioxide, dilute nitric acid, and organic acids. Only Krupp's V2A steel is resistant to organic acids and sulphur dioxide. Nickel steels are resistant to sulphuric acid. Marked variations are often observed in the same varieties of steel. The alloys are often highly corroded by boiling salt or acid solutions if air is brought into contact with the surface of the metal. Chromium-iron and chromium-nickel-iron alloys are far more resistant than ordinary steel to corrosion by salt solutions. The heat treatment before and after working the alloys has a marked effect upon the rust-resisting properties, and small variations may be very injurious. Even the polishing process influences the properties of the steel. The rust-resisting properties diminish if the steel be tarnished. Non-rusting steel and cast silicon alloys can be used in the construction of technical apparatus subjected to high temperatures and to the action of strongly acid gases, provided there is no condensation of the gases upon the polished metal surface; if, however, condensation is liable to take place, the only alloy suitable is V2A steel. A factor which must be taken into consideration with all steels, except V2A steel, as regards their suitability for the construction of chemical plant is whether the steel will be subjected to constant or only periodic attack by chemicals. (A. Brunner in *Vierteljahrschrift Naturforscher Gesellschaft*, Zürich, 1924, 69 [6], 1–79; *Chem. Zentr.*, 1925, 96, I, 1797–1798. Abstracted through *Journal of the Society of Chemical Industry*, vol. 44, no. 26, June 26, 1925, p. B452)

"Flakes" or "Hair Cracks" in Chromium Steel—"Shattered Zones" and "Transverse Fissures" in Rails

A tabulation of previous papers on flakes shows they have been found mainly in alloy steels and are considered to be actual cracks, but widely different causes have been suggested. Flakes have been found in heavy forged rings and bars of 1 per cent carbon, 1.5 per cent chromium steel made at the Gothenburg Works of the S. K. F. Co., being more frequent in larger forgings. At the Horfors Works no relation was found between the presence of flakes and any factor in the melting practice; and it was found that flakes could be removed by further rolling. On the hypothesis that flakes were formed during the last hot-working operation, numerous tests, described in detail, were carried out, to investigate the influence of different factors in forging. Unusually clean steel was used. Flakes were detected by treating a magnetized polished section of the sample with steel dust in gasoline, the dust then collecting on the cracks. The results were conflicting. Further experiments showed that flakes, when present, invariably formed during cooling after the final hot-working operation, and that their formation was prevented by retarded cooling. The defect may therefore be designated as cooling cracks. Flakes probably form after the Ar_1 transformation, and they mainly follow the austenite grain boundaries. If the steel is slowly cooled from too high a temperature, the cementite network is too coarse and annealing is hindered. Air cooling to about 750 deg., and then partial covering with charcoal, was found effective in eliminating flakes without marked interference with the annealing. Howard's discussion of shattered zones in rails is quoted as explaining the mechanism of the formation of flakes by cooling stresses. Large dimensions constitute an important factor promoting the formation of flakes. Inclusions and smaller heat conductivity due to chemical composition are contributory causes. So-called "silver streaks" in alloy steels are believed to be flakes that have been elongated in subsequent hot working without welding together. The literature on transverse fissures in rails is reviewed. The shattered zones found in rails are probably cooling cracks, and identical with flakes. Howard's views, rather than Dudley's, are supported by Hultgren. Rails could be made free from internal cracks by proper cooling, and might then not develop transverse fissures in track. Numerous references are given. (A. Hultgren in *Journal of the Iron and Steel Institute*, advance proof, no. 8, May, 1925, 36 pp. Abstracted through *Chemical Abstracts*, vol. 19, no. 13, July 10, 1925, p. 2016, *e*)

FUELS AND FIRING (See also Railroad Engineering: Fuel and Its Combustion on Locomotives: Testing and Measurements: The Degree Day)

The Calorific Value of Coal

Important work is likely to be done by the Joint Committee of the Institutions of Civil, Mechanical, and Electrical Engineers which has been formed to investigate the many problems associated with the testing of steam plants, with the object of devising standard methods of testing which will enable comparative tests of real value to be made as to the efficiency of different plants. One direction in which work of this description is particularly needed is in regard to the method of determining the calorific value of coal, there being at present no uniform method in use. The result is that comparisons made between various plants in connection with fuel statistics are not reliable, because it is not known how the calorific value of the coal has been determined. An attempt was made to prepare a formula for moisture, and the matter was taken up by a Committee of the British Engineering Standards Association. Since then the Joint Committee referred to above has been appointed, and the program which has been arranged provides for the whole of the problems involved to be carried out in a very thorough manner by the leading scientists and engineers in England. The Committee, of course, is a long way yet from having completed its labors, which are likely to last a long time (Editorial in *Mechanical World*, vol. 78, no. 2009, July 3, 1925, p. 2, *g*)

Inorganic Constituents of Coal

The author classifies the mineral constituents of coal according to their origin into seven classes. The classification is, however, only of theoretical interest.

According to Stopes and later investigators, bituminous coal consists substantially of four morphologically distinct and recognizable constituents, to which the names of fusain, durain, clarain, and vitrain have been given. (Compare Mechanical Engineering, vol. 42, no. 7, July, 1920, p. 405.)

It has been found, also, that these four constituents vary not only in regard to their total percentage of mineral matter but also in respect to the chemical composition of that mineral matter, as shown by a table in the original article.

From the data assembled it is possible to suggest tentatively the approximate distribution of mineral matter in bituminous coal. Broadly, mineral matter is distributed over the coal constituents as follows:

1 The inherent plant ash is contained in clarain and vitrain (bright coal). Some of it may have been abstracted, while some water-soluble salts and suspended clay have been trapped in these during "coalification." The gel-like structure or vitrain makes for great homogeneity, not only of its organic but also of its inorganic elements. The ash percentage in clarain is slightly higher, no doubt due to its comprising morphologically unaltered cuticles, spore exines, etc. which have retained their own original ashes.

Clarain and vitrain are relatively free from inorganic matter, yielding not more than 1.1 to 1.3 per cent of ash. This ash consists largely (65–70 per cent) of water-soluble salts, and a portion at least of the acid-insoluble matter (10–17 per cent) is trapped detrital material not strictly belonging to these groups.

2 Durain (dull coal) yields about 6 to 7 per cent of ash, largely composed of clay substance. Its acid-insoluble portion is preponderant. Durain ash is representative of the rock detritus, such as clay, which in the form of mud lodged between and within the plant fragments and small entities forming this deposit. It is an open question whether its hardness and higher specific gravity are due to the mineral admixture, or whether these properties must be ascribed to the kind of organic raw material and the conditions of its conversion into coal.

3 The ash of fusain ("mineral charcoal" or "mother-of-coal") is without doubt due to infiltration of "hard" water and subsequent deposition in this highly absorptive cellular material. This is borne out by the large proportion of the acid-soluble fraction, both in the original fusain and in its ash ($CaCO_3$, $FeCO_3$), and the wide range of its ash percentage from 4 per cent to 30 per cent. This indicates the element of chance as to whether an individual particle of fusain happened to be reached by the percolating stream of water. Again, the varying ash composition points to the possibility of such water having been of either calcareous or chalybeate (containing iron) character or having carried more or less clay in suspension.

4 The deposition of salts from aqueous solution in fusain is closely bound up with the segregation of the same substance in the cracks of the coal seams in the form of cleat and partings. They generally consist of the carbonates of calcium or iron and may assume the crystalline structure of calcite or its substituted forms (ankerites). These minerals, like the infiltration in fusain, are due to the decomposition of calcium or ferrous hydrogen carbonate from water according to whether this had previously passed through calcareous or ferruginous strata.

5 The analogy of cleat and partings with the fusain deposits extends to their containing similar amounts of insoluble matter. Much of this is clay substance filtered out or precipitated from suspension or colloidal solution in the carrying water.

6 A portion of the insoluble matter consists of iron sulphide.

The method for investigating the mineral constituents of coal is next discussed. The presence of the mineral matter has an important significance both on the mining of coal and its burning. As a result of the study of the inorganic constituents of coal an entirely new method of mining was evolved which is being tested in mining practice. This is based on a discovery that coal, when treated with acid of a concentration low enough to avoid an attack on the coal substance proper, loses its cohesion and can be crushed by the pressure of the fingers. This led to certain processes, among others the use of sulphur dioxide on masses of coal. Long bore holes of small diameter are drilled into the coal veins and a slow current of sulphur dioxide is passed into them at atmospheric pressure. The acid effects a distinct loosening of the mass and facilitates hewing work. It would appear that there is an entire absence of irritation or discomfort from gas escaping into the mine air, as any traces of sulphur dioxide not neutralized are absorbed by the coal and thus made innocuous. The quantity required is of the order of 2 lb. of sulphur dioxide per ton of coal, and it can be produced at a cost comparing favorably with that of explosives or mechanical aids to the getting of coal.

The coal-washing process is said by the author to be full of unsolved problems which require a closer study of the inorganic constituents and their behavior, particularly from the point of view of physical and colloidal chemistry.

The mineral constituents of coal also influence the process of combustion in various ways. Carbonization is subject to the catalytic influence of certain inorganic compounds, and hence the composition of the ash has an important bearing on this phase of the burning process. The importance of the fusibility of the ash is fully realized and of course the melting point of the ash depends upon its chemical composition. In fact, it can be calculated from the latter, at least to a certain extent.

The development of powdered-fuel firing calls for special attention to be given to the inorganic part of the coal. If a large tonnage of coal having ash contents as they are current today is burned in powdered-fuel installations, the emission of hundreds of thousands of tons of light ash would quickly substitute a dust menace for the vanishing smoke nuisance. In both producer-gas plants and carbonization in coke ovens and gas retorts the effect of the inorganic compounds is of great importance. In fact, laboratory work along this line seems to indicate that the present empirical processes of carbonization will have to be modified or replaced by methods which will permit influencing equilibria at will toward solid, liquid, or gaseous products and enable us to impart to them the commercially most desirable properties. (R. Lessing in a paper read before the Birmingham Section S. C. I. on Mar. 31, 1925. Abstracted through *Journal of the Society of Chemical Industry*, vol. 44, no. 24, June 12, 1925, pp. 277T–283T, *c*)

INTERNAL-COMBUSTION ENGINEERING

New Single-Acting and Double-Acting Krupp Engines

Description of new Krupp engines, the design of which is said to be based on the experience gained with the 12,000-shaft-hp. double-acting engine which was to have been used for driving war vessels. The merchant-marine Diesel engines differ, however, from the 12,000-hp. experimental engine in many important points. Thus far only two have been built—one single- and the other double-acting.

Both engines are of the two-cycle type. Below the cylinder head is a water-cooled shield which is attached to the cylinder only by the water connections. In this way the shield is free of all mechanical stresses and takes only the heat stresses.

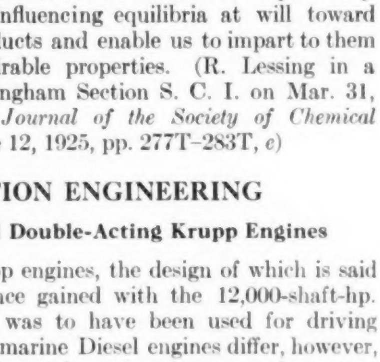


Fig. 2 Indicator Card of the Single-Acting Krupp Merchant-Marine Diesel Engine at Starting
(1 kg-cm.² = 14.22 lb. per sq. in.)

The usual arrangement of scavenge-pump drive is used, an arm on the crosshead carrying both the piston cooling-water pipes and the piston rod of the double scavenge pump. It is claimed that this not only does not produce any tilting effect on the crosshead but that the load due to the scavenge pump serves to counter-balance partially the tipping effect of the connecting rod.

Starting air and fuel are admitted simultaneously to all four cylinders at once, with the result that a very large mean torque is obtained. An indicator card showing the increased starting effort obtained by injecting fuel while air is admitted is reproduced in Fig. 2. It is claimed that whereas with compressed air alone the starting mean effort at right angles to the crank arm, referred to piston area, is 151 lb. per sq. in., with simultaneous injection it is 180 lb. per sq. in. (*Motorship*, vol. 10, no. 7, July, 1925, p. 528, *d*)

Centrifugal Treatment of Diesel Fuel Oils

Two articles, the first dealing with the M. S. *California* of the United Steamship Co. of Copenhagen, and written by the chief engineer (unnamed) of the motorship, and the second dealing with the general subject and written by Lee Hinchman Clark, chief chemist of the Sharples Specialty Co.

The motorship *California* is one of the earliest built by Burmeister & Wain and considerable trouble was experienced from the cracking of cylinder heads, requiring numerous replacements. Lately the vessel was bunkered with a 50–50 mixture of boiler oil and solar oil, which led to a good deal of injection-valve trouble. Investigation

showed that sand worked up from the lower end of the fuel valve into the bushing of the fuel-valve stem, with the result that the valve stuck tight.

To take care of such eventualities a centrifuge was installed and piped up with the fuel-oil system. The quantity of sludge it takes out from the fuel-oil mixture is almost unbelievable. The chief engineer states that he gets about 650 lb. of sludge from the daily supply of fuel oil, which is about nine tons. Part of this is water, part is sand, and the remainder is an asphaltic mess.

According to the second article, by Mr. Clark, there are distinct advantages to the oil-engine user in being able to burn the heavier so-called boiler-fuel oils in oil engines, as they are sold at a lower price and, by volume, on which basis they are purchased, have a slightly greater heating value. Furthermore they are more readily available on the market. On the other hand, the boiler-fuel oils are less popular as Diesel-engine fuels because, among other things, of the presence of small quantities of impurities in the oils. They are also more difficult to burn, but this has been overcome in recent designs. Impurities can be removed by gravity sedimentation or filtration, but both of these are less efficient than centrifuging.

In the past few years a great number of oil-engine power plants have been equipped with centifugals for the purification of fuel before burning. No difficulty has been found in effecting very complete elimination of harmful suspended impurities at high capacities when working upon fuels about 18 deg. Baumé and above. With fuels below this gravity the centrifugals are not as effective as they are with lighter fuels.

In order to obtain an entirely satisfactory purification of oils, except those that are very light, it is necessary to heat them before centrifuging. The temperature employed for centrifugal operation depends on the viscosity of the oil and the extent to which this viscosity must be reduced before satisfactory results can be obtained by centrifugal treatment. Thus to centrifuge an oil with the gravity in the neighborhood of 30 deg. Baumé, a temperature of approximately 120 deg. fahr. is usually sufficient. For heavier oils higher temperatures must be used, but as a temperature of about 150 deg. fahr. is often very close to if not above the flash point of many of these oils, there is a danger of explosion from escaping fumes developed when heavy oils are heated. To overcome this difficulty the Sharples Specialty Co. has developed a centrifuge in which the centrifugal operation is carried out in a totally enclosed housing within which the centrifugal bowl rotates, so that oil may be centrifuged at temperatures well above the flash point.

Data are reported of tests on a Sharples centrifuge on the double-acting two-cycle Worthington single-cylinder test engine at the Snow Holly Works, from which it would appear that from 1.6 to 6.0 oz. of solid material were removed from each 100 gal. of oil by the centrifuge. (*Motorship*, vol. 10, no. 6, June, 1925; first article, p. 441, 1 fig.; second article, pp. 442–444, 2 figs., *dp*)

LUBRICATION

Cylinder and Engine Lubrication

Laboratory tests of an engine equipped with a Rushmore steam cooling system, the object being to study the effect upon dilution of high cylinder-wall temperatures. The results show that a sharp reduction in dilution occurs as the boiling temperature is reached, and that the amount of dilution at temperatures of 212 deg. fahr. or more is much less than would have been anticipated from the results obtained at temperatures below 212 deg. fahr. The author then points out that high cylinder temperatures reduce dilution to a negligible quantity without introducing any apparent disadvantages.

A very important point, which so far seems to have been generally overlooked, is that lubrication, in one extremely important phase, will be interfered with very seriously by eliminating dilution. Cold-room and cold-weather testing have shown that oil flow is extremely sluggish with cold oil. This was brought out very thoroughly in an article by Frank Jardine, entitled Scuffed Pistons Result from Cold Jacket and Lack of Oil. His investigations were started to discover just why aluminum-alloy pistons were liable to "freeze" under fairly light loads at low temperatures. He demonstrated conclusively that it was due to nothing but the absence of oil. The importance of this work does not appear to have been properly realized. The correct way to look at this research is to think of the alloy piston as being simply an indicator that would show when lubrication was absent. Under the same conditions, with pistons and other material, precisely the same lack of oil and, consequently, excessively rapid wear of piston rings and cylinders would occur. This the aluminum piston indicates at the time, whereas the iron piston does not.

At present it is not an overstatement to say that nearly all automobiles used in extremely cold weather are protected by dilution. If dilution is really eliminated without any modification of the lubrication system, an epidemic of mechanical failure will come with the first winter. Of course, this condition can be overcome by the use of a lubricant for winter work so light that its viscosity is the same as the present oils have with the normal amount of dilution; but assuming that the object of eliminating dilution is to maintain viscosity, then changing to a lighter lubricant would be utterly ridiculous. The fact is that the conventional lubricating systems are faulty fundamentally, in that they function properly only over too narrow a temperature range. What the pistons, the cylinders, and the bearings require is a constant volume of oil per revolution, irrespective of viscosity. (A. Ludlow Clayden in *Journal of the Society of Automotive Engineers*, vol. 17, no. 1, July, 1925, pp. 58–61, *e*)

MACHINE PARTS

Clutches and Transmission for the RS-1 Semi-Rigid Airship

This airship, which is now nearing completion at the Goodyear Tire & Rubber Co., Akron, Ohio, is to have two power cars, each equipped with two Liberty 12-A engines driving a $17^1/_2$-ft. propeller through reduction gears. Both the clutches and transmission have been designed by the Twin Disc Clutch Co. of Racine, Wis. The clutch is operated by toggle levers and has no springs with the exception of six light ones which separate the plates when the clutch is disengaged. It is said that this construction entirely eliminates any possible end thrust against the crankshaft bearings. The clutch will stay in either the engaged or disengaged position, with no chance for the clutch to engage when engagement is not desired.

One of the features of the clutch is the simple and quick means for adjustment, which is by an adjusting pin. It is only necessary to pull out this pin to a stop and then turn one-twelfth turn or more as may be necessary. All the clutch plates except the two outer, which are duralumin forgings, are made of the highest-grade saw-blade steel with copper-asbestos woven disks riveted to the driven disks. (*Aviation*, vol. 19, no. 1, July 6, 1925, pp. 12–13, 2 figs., *d*)

MACHINE SHOP

Flow and Rupture of Metals During Cutting

Report of the Cutting Tools Research Committee of the Institution of Mechanical Engineers. The main object of the investigation was to study the deformation and flow of metal under the action of a simple cutting tool, and more particularly to observe the manner of separation by which the chip is formed. This object was pursued by studying the internal structure of chips in process of formation under different cutting conditions and in different stages of development. In order to make the observation as simple and direct as possible and to reduce the number of independent variables entering into the problem, the cuts studied in the present investigation were all made on the edges of revolving disks. The experimental cutting operations were performed with ordinary carbon-steel parting tools, and the only angles which were varied in the course of the investigation were those of top rake and front clearance. No lubricant was used.

The study of the cutting process which is described in the present report is essentially in a preliminary and tentative nature. The method of study, namely, the microscopic examination of cross-sections taken through chips in process of formation, has, however, led to a series of interesting observations which seem to afford a new insight into the mechanism of cutting-tool action. By using

a simple type of cut and varying only two factors, namely, depth of cut and top-rake angle, certain definite regularities of behavior have become manifest. These may be summarized by saying that, according to the conditions of cutting, the separated metal takes the form of three distinct types of cutting, or "chip." In the first of these types, which has been termed the "tear" type, rupture of the metal occurs by the formation, well in advance of the nose of the tool, of a tear or crack tending to run inward from the periphery of the stock. Since such a tear cannot obviously, progress very far, a succession of fresh starts are made by the tool, and the surface of the work is left in the form of rough projections, each of which probably corresponds to successive shearing of the chip. When the conditions are changed by either reducing the depth of cut or increasing the top-rake angle, or both, the ultimate result is the formation of the third type of chip, which has been called the "flow" type, but under intermediate conditions an intermediate type of chip is formed which has been termed the "shear" type, since it is mainly formed by a process of shearing on a plane making an angle of roughly 30 deg. with the direction of motion of the tool. The change from one of these types to the next is gradual, but an attempt has been made to determine in a rough quantitative manner the relations of the two variable factors to the various characteristics of the cut in regard to the nature of the chip, the roughness of the surface produced, and the average depth of cut made as compared with the intended depth. In the formation of the "flow" type of chip the principal movement occurs, without actual fracture, in a direction at right angles to this plane of shear, and usually takes the form of a continuous spiral or ribbon; this is definitely associated with the smoothest-cut surface and the closest agreement between intended and average depth of cut.

So far as the present investigation goes, it indicates that the best results in cutting in regard to the removal of the maximum amount of metal per unit distance of tool travel, the least irregularity of surface, the closest agreement between intended and actual depth of cut, and the minimum wear of tool, is obtained by using a top-rake angle a very little smaller than that at which the heavily deformed zone before the nose of the tool just disappears, in conjunction with the greatest depth of cut which still allows the formation of the "flow" type of chip.

If it can be assumed that the three types of chip which have been studied in this investigation retain their characteristics and mode of formation under different conditions of cutting, it will now be possible, by simply observing the type of chip produced, to determine the effects of varying the speed of cutting, the shape and nature of the tool, and the nature of the metal being cut, as well as the effects of different lubricants and other variables. It is clear, however, that so wide an application of the present methods entails much further detailed study. Dr. Walter Rosenhain and A. C. Sturney, both of the National Physical Laboratory, Teddington, England, in *Proceedings of the Institution of Mechanical Engineers*, no. 2, Jan., 1925, pp. 141–174, 28 figs., *eA*)

The Spray Method of Cleaning Machinery

In many places spray guns, similar to those used for painting, are now being used to supplant or supplement the work of the engine wiper and for cleaning all kinds of shop machinery. For cleaning electrical machinery gasoline is sometimes used as the cleaning agent, since it will remove the dirt without injury to the insulation. Probably the greatest objection to the use of gasoline is the fire hazard. Where injury to insulation is not a factor, the fire hazard is eliminated or at least greatly reduced by using an emulsion of water and gasoline. Dirt having a tar content will give way rapidly before a water and gasoline spray. For cleaning the running gear of locomotives, a mixture of water, kerosene, and soap is used. The practice of cleaning machinery in this way results in the saving of much labor and in cleaning out-of-the-way places effectively. It will probably be used more widely in the near future, and there remains the problem of determining the proper cleaning agent for each application. Different kinds of cleaners are needed, and the proper one must be selected for each different application. (Editorial in *Railway Mechanical Engineer*, vol. 99, no. 7, July, 1925, p. 403, *d*)

MACHINE TOOLS

The Kango Hammer

An electrically driven percussion tool invented by Professor Goldschmidt which depends fundamentally on the utilization of centrifugal force for the conversion of rotary into impulsive motion. It is stated that in a demonstration such operations as chipping steel, keyway cutting, shaping wood, dressing stone, and drilling brickwork were satisfactorily performed, the action and nature of impact being very nearly even. In fact, a chip from the edge of a mild steel plate equaled in smoothness that obtained from a good cut in the lathe. The hammer itself was run from an ordinary electric-lamp socket, the ammeter reading at no time exceeding 1.25 amperes. The tool weighs about 9 lb. It is not suitable in the present form for such heavy duty as riveting or the removal of a weld upset.

The principle on which the Kango hammer depends will be understood by the aid of Fig. 3, which shows the arrangement of the tool in section.

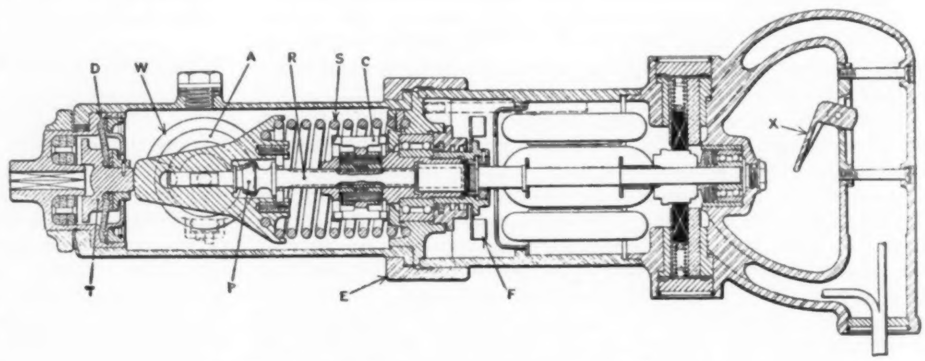


Fig. 3 General Arrangement of the Kango Hammer

The reciprocating and progressive effect, which amounts to a definite blow, is obtained by the rotation of the two heavily loaded bevel gearwheels, one of which is seen at A, placed facing each other and driven through the bevel pinion P in opposite rotational direction.

The weights of the loaded gears, one of which is shown at W, are in this way brought into juxtaposition at two points in their path of rotation per revolution, converting their centrifugal force into a sharp oscillating motion forming the blow, which is conveyed to the chisel or tool through a transmitter T held in position by an oil-retaining leather diaphragm D. The recoil is taken by a spiral spring S, which serves to limit the oscillation of the striking tool, absorb vibration, and to add impetus to the blow on the tool.

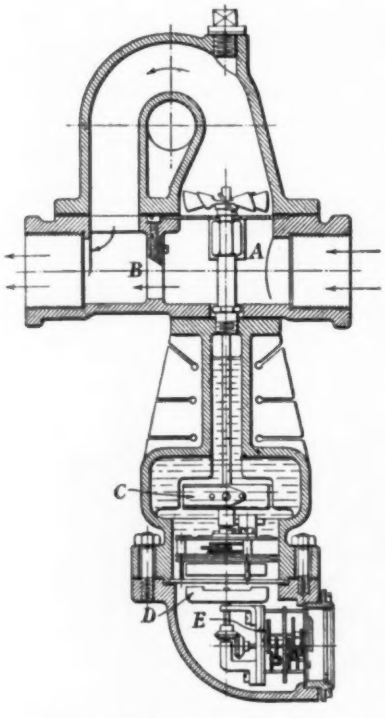
The most interesting point of this new principle is the part played by the speed of rotation of the weights and the effect on the oscillating motion at the hammer head. Variation of speed produces a harmonical variation in the number of blows, and as the speed is increased, blows of proportionately increased intensity to those obtained at one blow per revolution can be obtained at every second or third revolution. This point is being put into practice for larger hammers, which will be produced in due course. (*Machinery* (London), vol. 26, no. 666, July 2, 1925, pp. 435–436, 4 figs., *d*)

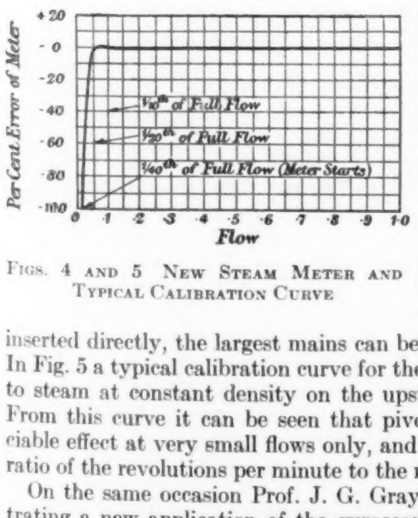
MEASURING APPARATUS

New Steam Meter—Monorail Car

In connection with the "conversazione" of the Institution of Civil Engineers in London, June 17, 1925, an exhibit was held. Among other things was shown a new form of steam meter exhibited by J. L. Hodgson and made by Geo. Kent, Ltd., of Luton (Figs. 4 and 5). A portion of the steam passing through the in-

strument is diverted by means of an orifice plate through nozzles S, by means of which it is caused to impinge on a turbine. The speed of this turbine is kept low by means of a damping fan C, which rotates in water, resulting in a minimum of wear on the turbine bearings. The retarding torque at any given speed due to this fan is made so great that any variation in the corresponding torque due to the turbine, caused by variations in the density of the steam in which it rotates, can be neglected. The turbine rotates at normal flows with a speed proportional to $Q/\sqrt{W}$, where Q is the weight of steam passing per second, and W is the density of the steam in pounds per cubic foot. From this it will be seen that a variation of 2 per cent in the density of the steam will involve a correction of only 1 per cent in the flow reading. Besides reducing wear on the pivots, the use of the damping fan increases the range of the meter, since it increases the relative velocity between the impinging steam and the turbine. The counter E is mounted in a compartment below the main, a magnetic drive D from the turbine being employed so that leakage of steam or water into this compartment is impossible. Radiating fins assist in keeping the counter cool. The maximum flow which the meter will measure can be altered by changing the orifice plate B; and by fitting a blank plate in place of this orifice plate, and putting the meter in a bypass in a large main into which a suitable orifice plate is inserted directly, the largest mains can be metered at a small cost. In Fig. 5 a typical calibration curve for the meter is shown, reduced to steam at constant density on the upstream side of the meter. From this curve it can be seen that pivot friction has an appreciable effect at very small flows only, and that at normal rates the ratio of the revolutions per minute to the rate of flow is constant.





FIGS. 4 AND 5 NEW STEAM METER AND TYPICAL CALIBRATION CURVE

On the same occasion Prof. J. G. Gray exhibited a model illustrating a new application of the gyroscopic principles to a monorail car. In this mechanism the car was articulated, the front portion carrying the gyroscope with the wheel vertical and in the plane of motion, the rear portion carrying the tractive element. When the car inclined to one side the articulation resulted in the axes of the two portions getting out of line, and in this position the rear portion in pushing the front portion tended to accelerate the precession, this resulting in a stabilizing couple being produced by the action of the gyroscope. (*Engineering*, vol. 119, no. 3103, June 19, 1925, pp. 770–771, 2 figs., *d*)

MECHANICS

Determination of the Stresses in a Beam by Means of the Principle of Least Work

SAINT-VENANT has given a solution of the problem comprising, in particular, the determination of displacements, strains and stresses in cylindrical beams fixed at one end and loaded at the other. When the displacements are given it is an easy matter by means of the general equations of elastic equilibrium and by a simple differentiation to determine the forces which produce them. The inverse problem to determine the displacement when the forces are given is of far more practical importance, but had not been solved in a general way when Saint-Venant took up the problem on account of the difficulty of integrating the equations in which the displacements enter, and of determining the functions and arbitrary constants of integration in such a manner as to satisfy the conditions in the various cases.

Saint-Venant solved the problem by making certain assumptions. Later on, Clebsch solved the problem for a beam fixed at one end and loaded in any manner at the other end, including bending, torsion, and tension. He made two out of three assumptions formerly made by Saint-Venant.

The present author follows the inverse method. All the external forces as well as their distribution over the end sections are assumed to be known, while none of the displacements are given. By means of the principle of least work, the stresses are determined and from this the displacements can be found. By using this principle and the calculus of variations the author makes it possible to embody all the conditions together with the expression for the internal or elastic work of deformation in one integral, by the variation of which the unknown stresses can be determined directly. It is claimed that the solution, being based on the fundamental principle of least work, gives assurance that the state of stress to which it leads is one to which the beam must tend, even though it may not in practice conform to it near the point of application of the external forces. In other words, it gives the ideal or natural stress distribution. The author applies his method to several problems, in particular, the state of stress in a cylindrical or prismatic beam of any form of section and of isotropic material fixed at one end, while at the terminal section of the free end is applied a system of forces. The axis of the beam is supposed to be horizontal. No external forces act on the lateral bounding surfaces and the action of gravity is neglected. The solution is given by a process of mathematical reasoning not suitable for abstracting. Expressions are found for the elastic work of deformation, equations of conditions, longitudinal stress, etc. It is shown that by means of the principle of least work it is possible to determine the stresses when the external forces acting on the beam are known and to do it without making any *a priori* assumptions regarding the displacements as is done in Saint-Venant's method. Once the stresses are determined the displacement can be found as explained in Love's Theory of Elasticity (3rd edition, p. 338).

It is a not uncommon fallacy to believe that the ordinary formula for bending depends in all cases for its correctness on the assumption that plane cross-sections of a beam remain plane and normal to the longitudinal filaments after bending. This condition is fulfilled only in pure bending where the beam takes a circular form, but as shown by Saint-Venant and as brought out in the present paper, the ordinary bending formula is rigorously true also in unequal bending, provided the shear force is constant, as in a cantilever beam loaded at the end by a transverse force. In that case the cross-sections do not remain plane, but curve over and incline equally toward the filaments, so that the proportionality of the longitudinal stresses expressed by the ordinary formula is strictly maintained. (Prof. Wm. Hovgaard, Massachusetts Institute of Technology, in *Journal of Mathematics and Physics, Massachusetts Institute of Technology*, vol. 4, no. 2, April, 1925, and also in *Bulletin of the Massachusetts Institute of Technology*, vol. 60, no. 64, April, 1925, pp. 103–129, 2 figs., *tm*)

MOTOR-CAR ENGINEERING (See also Testing and Measurements: Automobile-Noise Measurement: Transportation: Street Congestion and Motor-Sales Saturation Point)

Six-Cylinder Bentley Car

THIS car is of interest because of certain features in the design of the engine and chassis. The camshaft drive is located in a tunnel, which materially increases the length of the engine and

gives it a more pleasing appearance. Furthermore, convention has been disregarded and the normal train of gears discarded in favor of three connecting rods. From the crankshaft a gearwheel drives another fabric gear, which in its turn drives a small three-throw crankshaft of which each crankpin is provided with the equivalent of a small big-end bearing. On the tail end of the camshaft is a similar three-throw crankshaft also provided with three big-end bearings, and the two sets of bearings are each connected by a pair of stout tubes. The result is that the drive is transmitted regularly and evenly to the camshaft at half engine speed, and the only point where there might be noise is in the reduction gearing.

The drive was not easy to design. It is necessary, for example, to allow for any difference in the distance between the camshaft and crankshaft centers to permit the gear wheels to be fully meshed and to allow for the expansion of the engine when heated. This has been done by holding the upper bearing for each pair of rods to the rods between powerful coil springs which balance each other. If the bearing on the camshaft has to be moved upward slightly, it compresses two springs by the amount by which the other two springs are released. The bearings for the rods are of aluminum and the small crankshafts are hardened.

An entirely new type of clutch has been adopted. The clutch fork has a beam equalizer as before, but, in place of rollers, two small links give a straight push at the clutch collar, which in turn operates long levers, withdrawing one of the driving disks from the fabric-faced driven disk. The levers can be adjusted so that each bears an equal amount of the load. The driven disk is of duralumin with a fabric face each side to avoid overheating the disk if the clutch slips. The duralumin is bolted to the clutch shaft through two fabric star-shaped pieces, the disk itself being centered on the clutch shaft in such a manner that part of the drive can be taken direct on the disk as well as through the fabric, and yet the disk allowed a certain freedom of movement. Six powerful coil springs grip the disks. Even the self-aligning spigot ball race has a wick lubricator, and, should any oil escape into the clutch, it is caught in gutters and transferred outside. (*The Autocar*, vol. 54, no. 1548, June 19, 1925, pp. 1075–1079, illustrated, *d*)

Unconventional Transmissions

A SYMPOSIUM of two articles, the first by T. M. Heldt on a general technical study of the work that has been done in the last few years in an effort to improve existing methods of engine-torque conversion, and read at the Summer Meeting of the Society of Automotive Engineers; and the second, by W. F. Bradley, dealing with the DeLavaud automatic transmission described in MECHANICAL ENGINEERING, vol. 46, no. 1, January, 1924, p. 46.

Mr. Heldt gives a chart showing two sets of curves which bring out the conditions of operation under which the thermal efficiency or the fuel economy is the highest. From this it would appear that if we had a continuously variable transmission the engine could be operated at any point along the corresponding "equal-horsepower curve," i.e., the engine could be operated either at high speed and low torque or at low speed and high torque, or at some intermediate point. As we follow an equal-horsepower line in the direction from high to low speed, i.e., from right to left, we cross or approach lines of lower specific fuel consumption, or high fuel economy. The improvement in fuel economy with decrease in speed is particularly pronounced at high horsepower, but in order to be able to take full advantage of the increase of fuel economy with decrease in speed for equal horsepower, a continuously variable gear is needed instead of the one used today, which gives only two, three, or four changes.

A number of such continuously variable gears have been designed, but the vast majority of them are not suitable for commercial operation. The author describes a number of methods such as transmission by truncated-cone pulleys and belt, mechanical drag devices, magnetic drag and hydraulic drag (for example, the Radcliffe hydraulic drag type, etc.).

The simplest way to obtain continuous variability of the ratio consists in the use of a variable-throw pump, and the most familiar type of such pumps is that in which the pump plungers are operated by means of a swash plate, i.e., a revolving plate set at an angle to its shaft or axis of rotation (hydraulic drive made by the Waterbury Tool Co., Waterbury, Conn. Compare MECHANICAL ENGINEERING, vol. 44, p. 666, and vol. 46, p. 493). The article is to be continued.

Mr. Bradley describes the application of the DeLavaud transmission to the 10-hp. sleeve-valve Voisin car and his article is based on a test in which a total of 2164 miles was covered under his direct supervision.

From these tests it would appear that the DeLavaud transmission has certain important advantages, such as a 15 to 26 per cent decrease in gasoline consumption and a 50 per cent decrease in oil consumption, more rapid acceleration, simplified control, and a wide range of speed varying from 3 to 50 m.p.h. with throttle control only. The use of the DeLavaud transmission embodies a coaster axle which gives perfectly free running. (*Automotive Industries*, vol. 52, no. 25, June 18, 1925. Article by P. M. Heldt, pp. 1052–1056, 8 figs., and article by W. F. Bradley, pp. 1058–1059, 1 fig., *dc*)

The Berk Supercharger

DESCRIPTION of the supercharger fitted to the Austin "Seven," differing from other superchargers in having three blades. This supercharger is therefore a good deal like the spur-gear oil pump in

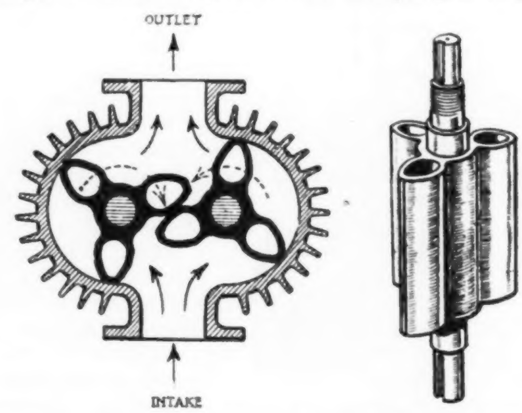


FIG. 6 THE BERK SUPERCHARGER WITH 3-BLADED ROTOR

its action. The blades, machined in one piece with their shaft, are hollow and are shaped as the teeth of a gearwheel, two of the three-bladed rotors running together in a casing; but though the blades almost touch, they do not drive each other like gears, but are driven by separate gears of ordinary type in a compartment in the supercharger case, the separate gears being driven through some form of flexible coupling from the crankshaft of the engine and at crankshaft speed or slightly higher. As the clearance between the blades of one rotor and those of the other is a matter of a few thousandths of an inch, the gearing between the two rotors has to be very accurately meshed. Therefore a boss is keyed to the rotor shafts and drilled with a number of accurate holes in relation to the rotor blades. A gearwheel is then pushed over the boss, making a good fit, and bolted in position, the bolt holes in the gear having been located accurately from its teeth. By this means the relation of the blades on one rotor to those on the other can be determined accurately.

Great care has to be taken to set the clearance between the tip of the supercharger blades and the external casing, partly because centrifugal force has, in certain instances, distorted the blades, and partly in order to prevent the air or gas which the supercharger is pumping from escaping; and for the same reason the clearance between the blades of one rotor and the boss of the other has to be set with care. Both rotors run on roller bearings with ball thrust races provided with lubricant from the oil pump of the engine, and the blades themselves are sometimes lubricated by allowing oil to enter with the air or gas according to whether the supercharger is forcing air through the carburetor or sucking gas from it.

Another point that has to be considered is that the supercharger becomes hot, for which reason fins are cast on the casing, and even the air or gas passing from the supercharger has to go through an intermediate cooler in the form of a radiator before entering the engine. It will thus be seen that there are a number of problems to be tackled in connection with superchargers.

The casing which contains the rotor and the intake and outlet pipes is one casting, one of its end covers carrying the gears and both taking the bearings for the rotor shaft. These bearings must be accurately located, therefore the Berk patents cover a spigot end for the main casing which registers with recesses in the end covers and locates them accurately, the bolts holding the covers in position merely gripping the flange of the end cover to the flange of the main casing. (*The Autocar*, vol. 54, no. 1545, May 29, 1925, p. 936, 3 figs., *d*)

POWER-PLANT ENGINEERING

Power Facilities at New Union Station in Chicago

At the new Union Station in Chicago to be operated by the Pennsylvania, Burlington, St. Paul, and Alton railroads, enormous installations have been provided to take care of the heating and ventilation requirements of the plant. An idea of the magnitude of the ventilating system is offered by the fact that the complete fan installation has an aggregate capacity of 1,220,000 cu. ft. of air per min. There are 35 fans in the station proper and five in the mail building, the largest of which has a capacity of 67,000 cu. ft. of air per min. The ventilation in the station structure embraces both plenum and exhaust systems. Eleven fans have been provided for exhausting smoke under the viaducts at Jackson Boulevard and Adams Street.

The heating system in the station proper, which is primarily one of the direct radiation type, is supplemented in large part by a tempering of the large volume of air driven through the ventilating system. Hot-water heat has been adopted for the mail building.

The supply of the energy required in the terminal, not only in the form of steam for heating but also the power for the operation of the pumps, fans, compressors, etc. is large and imposed an important problem with respect to the power supply, no small portion of which concerns the electric current for the lights, signals, and the operation of the mechanical equipment. A careful investigation by the station company led to the conclusion that the greatest economy and reliability of service would be afforded by the purchase of electrical power from the local public utility, the Commonwealth Edison Company, rather than by constructing and operating its own central station plant. Consequently the power plant provided by the station company is designed solely to provide steam for heating the various buildings, to supply hot water for domestic use, steam-heat supplies for cars and for the operation of air compressors to supply compressed air for the operation of the switches in the interlocking system and for the charging of trains. (Part of an article by Walter S. Lacher, describing the Chicago Union Station, in *Railway Age*, vol. 79, no. 1, July 4, 1925, pp. 7–28, illustrated, *d*)

Concentration in Boilers

While this subject is of great importance, certain features of it appear to have been seriously neglected. The two best previous studies to which the author refers are by C. W. Foulk, of Ohio State University, published in *Industrial and Engineering Chemistry*, vol. 16, p. 1121 (compare abstract in Mechanical Engineering, vol. 47, no. 1, Jan., 1925, p. 49), and by R. E. Hall, of Pittsburgh, in Mechanical Engineering, vol. 46, Mid-November issue, 1924, p. 810.

According to Mr. Foulk, the point at which foaming will occur is dependent upon the relation of the minute solids to the soluble salts in the boiler water. In his opinion, of even more importance is the concentration of the actual materials which increase the surface tension to a point at which bubbles become more or less permanent. Each set of conditions gives different points above which concentrations should not be carried.

It is impossible to prevent concentration as long as soluble impurities are carried into the boiler. According to an editorial in *Power* by F. R. Low (Mem. A.S.M.E.), "The impurities removed from a boiler over a long period most equal those brought in by the feedwater."

Mr. Hall in his paper in Mechanical Engineering recommends, from a scale-forming standpoint with his particular treatment, that the concentration be kept below 2000 parts per million, which is only 118 grains per gallon. This may be correct from a treatment standpoint, but the boiler operator must balance the slight increase in scale which might be formed according to Mr. Hall's theory against the actual material loss occasioned by the excessive blowdown necessary with Pittsburgh waters. For instance, a 20-grain or even 30-grain water is not uncommon in this district. To keep the concentration down to 117 grains per gallon with a 30-grain water would require a blowing down of roughly 25 per cent, or a loss of heat from this practice of roughly 6 per cent. If, however, this particular plant could be operated with a concentration of 600 grains instead of 117, the blowdown would be 5 per cent instead of 25 per cent, and the heat loss only 1.2 per cent—a direct saving of 4.8 per cent in efficiency—and such a saving will pay for a lot of labor for boiler cleaning.

To see what this means in terms of frequency and amount of blowdowns, take, for instance, a cross-drum boiler with a drum 54 in. in diameter and about 15 ft. long. A blowdown of 4 in. would amount to roughly 170 gal. If the output of the boiler is 1000 boiler hp. and the quality of the feedwater is 15 grains per gallon, we are introducing into the boiler 54,000 grains of impurities per hour. If, for instance, the point at which this boiler starts to foam violently is at 300 grains per gallon, it will be necessary in the interest of safety to blow this down when the concentration reaches a point of, say, 250 grains. Each blowdown will therefore take out of the boiler 250 × 170 or 42,500 grains total, which means that in this particular case the boiler would have to be blown down 4 in. every 45 min. in order to keep the concentration from rising above 250 grains if no moisture passed out with the steam.

With 1.5 per cent going out with the steam, this would take out 54 gallons of water per hour at a concentration of 250 grains per gallon, or 13,500 grains per hour, leaving 40,500 to be removed through the blow-off valve, or a blowdown about once each hour. In a case like this, if the water tender should forget himself and let the boiler run, say, for two hours without blowing down, long before this time the critical point would have been passed and the boiler would be throwing water violently into the steam lines. It is conditions such as these that have been responsible for a great many wrecked cylinder heads. If we had to keep the concentration below 117 grains per gallon with 1.5 per cent moisture, the steam line would take 6350 grains per hour. Each blowdown would take 20,000 grains and the boiler must be blown down about every 25 min.

The author proceeds to the consideration of the reactions involved in feedwater treatment, in particular, the soda-ash process, and comes to the conclusion that such treatment increases rather than decreases the impurities and to this extent increases the concentration in the boilers. However, the impurities that are left are practically non-scale-forming materials, so that the treatment does what it was designed to do—prevent scale; but actually the boiler concentrations using treated water will ordinarily be greater than with raw water. The treatment with zeolite produces similar results.

From this the author proceeds to a discussion of methods for determining the concentration in a boiler, particularly to the very important matter of the method of taking representative samples. In checking the accuracy of concentration it is advisable to balance up the totals; in other words, to make up for the boilers a "dirt balance" in the same way that a heat balance is made up to prove the accuracy of a boiler test. The author shows how to do this.

The author discusses also throttling calorimeters, calling particular attention to the necessity of determining the "normal" of each instrument as provided in the A.S.M.E. Power Test Code of The American Society of Mechanical Engineers. To show the necessity of this, the author gives particulars of a series of tests made with exceptional care and laboratory precision on four different types of throttling calorimeters, each type tested with various sizes of orifices. These orifices from $^3/_8$ in. down to $^1/_{16}$ in. in diameter. Some of the calorimeters were lagged with asbestos, some steam-jacketed, and some treated by both methods. The normal of these various types of sizes—in other words, the number of degrees which must be added to the lower reading to correct for the proportional effect of radiation, velocity impact, etc.—varied all the way from 1.6 up to 76 deg. fahr. In running a test in the field the normal should be determined both before and

after the test; or, if only one normal test is taken, it should be directly after the test, for a piece of pipe scale lodging in the orifice might easily change the normal anywhere from 5 to 12 deg.

In the discussion which followed a good deal of attention was devoted to the barium treatment. Some said it was too expensive, while others said that in a certain case barium treatment had been abandoned because while it was used tubes got dirtied by oil and ultimately burned out. K. von Eltz, manager of the Reisert Automatic Water Purifying Co., New York, explained why the barium treatment was supposed to be efficient and described the apparatus that has to be used with it. Another speaker explained that where the barium treatment is used mineral and lubricating oils should be eliminated before reaching the boiler. (Grant D. Bradshaw, of Andrews-Bradshaw Co., Pittsburgh, Pa., in *Proceedings of the Engineers' Society of Western Pennsylvania*, vol. 41, no. 4, May, 1925, original articles pp. 105–118, 4 figs. and discussion pp. 119–132, 1 fig., *pA*)

RAILROAD ENGINEERING (See also Engineering Materials: Flakes or Hair Cracks in Chromium Steel: Measuring Apparatus: New Steam Meter—Monorail Car)

Garratt Locomotive for the London and North-Eastern Railway

DESCRIPTION of a Garratt locomotive, which is not only the first engine of the type built for a British railway, but the heaviest yet built by Beyer, Peacock & Co., Ltd., and the first to have two three-cylinder engines. Each unit consists of a 2-8-0 engine, of which the cylinders, valve motion, etc. are interchangeable with those of the London and North-Eastern Railway's 2-8-0 three-cylinder coal engine. It is to be used as a banking engine on the Worsboro' Branch, between Wath and Peniston, which is 7 miles long, all on a rising gradient, with 2 miles at 1 in 40 up. The trains weigh about 1000 tons, and are worked at present with two train engines in front and two or three banking engines behind. The Garratt will replace the banking engines. The boiler for this huge machine is 7 ft. in diameter, and the tractive effort is equal to that of two of the London and North-Eastern Railway 2-8-0 coal engines. For previous references to the Garratt locomotives, see MECHANICAL ENGINEERING, vol. 44, no. 10, Oct., 1922, p. 665. (*The Engineer*, vol. 139, no. 3626, June 26, 1925, p. 715, *d*)

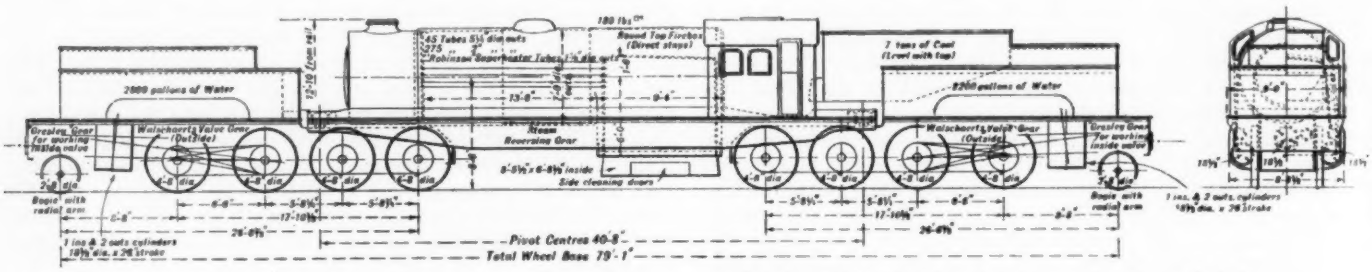


FIG. 7 GARRATT LOCOMOTIVE FOR THE LONDON AND NORTH-EASTERN RAILWAY

Exhaust-Steam Injectors

THE author starts by a theoretical explanation of the reasons why an exhaust-steam injector can deliver feedwater to a boiler against a higher pressure than the steam pressure, and from this proceeds to a description of the most recent type of these injectors brought out in Europe, namely, the one built by Alex Friedmann in Vienna, Austria. This injector is so arranged that with the opening of a single steam valve not only are the proper paths for water and live steam thrown open, but the exhaust-steam valve opens by the opening of the device and closes when it is closed. Furthermore, the throttle jet of live steam is simultaneously delivered to the injector, and in due time shut off. (Hans Deutsch in *Schweizerische Bauzeitung*, vol. 85, no. 24, June 13, 1925, pp. 301–303, 7 figs., *d*)

The Three-Cylinder High-Pressure Locomotive

THREE-CYLINDER high-pressure locomotives are coming into use in all parts of the world, and present many advantages over other types. The paper considered here reviews the gradual development of the three-cylinder locomotive from the prototype designed and built by Stephenson in 1846 at Newcastle.

In 1907 the use of this cylinder arrangement became more general to meet the greater demands for power, and at the same time comply with the limitation of load gages. The development has been made chiefly on the London and North-Eastern and on the German State Railways, the increased use of the arrangement having been assisted by the author's valve gear requiring two sets of motion only instead of three, as used on the earlier engines.

The paper gives particulars of comparative tests of 4-4-2 and 2-8-0 two- and three-cylinder engines. Particulars of the economy in coal and water shown by the three-cylinder engines are given, and these figures are confirmed by the results obtained in actual working. The paper then deals with the improved starting effort of three-cylinder locomotives, due to the more uniform turning moment. Actual records showing the fluctuations per revolution with two- and three-cylinder engines of the same type are reproduced from dynamometer-car records.

The increased mileage between repairs obtained with three-cylinder engines is then mentioned, and particulars of reduction in tire wear given. The steadier running of these engines at high speed is then considered, and the reduced hammer blow of three-cylinder engines compared with that of two-cylinder engines is brought out, showing that an increased static load on the rail could be permitted in the case of such three-cylinder locomotives.

The steadier vacuum in the smokebox with three-cylinder engines is noted, and particulars are given of experiments made to ascertain the actual conditions in the smokebox.

The arrangement of valve gear is briefly described to show how the third valve is operated from the outside valve gears by a suitable arrangement of levers.

The paper concludes with a summary of the advantages of the three-cylinder locomotive, and is illustrated by eight figures and accompanied by four appendices containing fourteen figures. (Herbert N. Gresley, in a paper before the Institution of Mechanical Engineers, July 7, 1925. Abstracted through the June, 1925, *Journal of The Institution of Mechanical Engineers*, pp. 138–139, *dg*)

Tests of Three-Cylinder Locomotive by the M. P. R. R.

DATA of tests made in the testing plant of the Pennsylvania Railroad at Altoona, Pa., of a three-cylinder locomotive recently built by the American Locomotive Co. for the Missouri Pacific Railroad. The locomotive is a Mikado 2-8-2 type with a straight-top-type boiler operating at 200 lb. pressure.

As originally delivered to the testing station the locomotive did not fire well on heavy loads and did not steam freely. Some changes were then made, such as removing the basket bridge and adding Goodfellow projections to the 7-in. diameter nozzle, and these, with the changed stack, were found to make the steaming and draft conditions satisfactory.

In previous tests of two-cylinder locomotives it was found that the fore-and-aft motion or vibration due to the unbalanced recipro-

cating parts became so severe at a speed even below 200 r.p.m. that additional balance weights were needed in the wheels for the test-plant operation, and it has been customary to add sufficient weights to balance completely all the reciprocating weights. With the three-cylinder locomotive no additional balance weights were applied and a speed of 235 r.p.m. was reached before the fore-and-aft vibration became violent enough to endanger the mechanism of the dynamometer. It was concluded that without special counter-balancing a two-cylinder locomotive could be operated safely at a speed of 180 r.p.m. and a three-cylinder locomotive at a speed of 240 r.p.m., or at approximately one-third greater speed. Complete data of tests and indicator diagrams are given in the original article. (*Railway Age*, vol. 78, no. 30, June 27, 1925, pp. 1623–1628, 13 figs., *e*)

The Modified Fairlie Locomotive

Description of the unit built by the North British Locomotive Co., Ltd., Glasgow, for the South African Government Railways. The Fairlie locomotive is of the articulated type, Fig. 8, and in

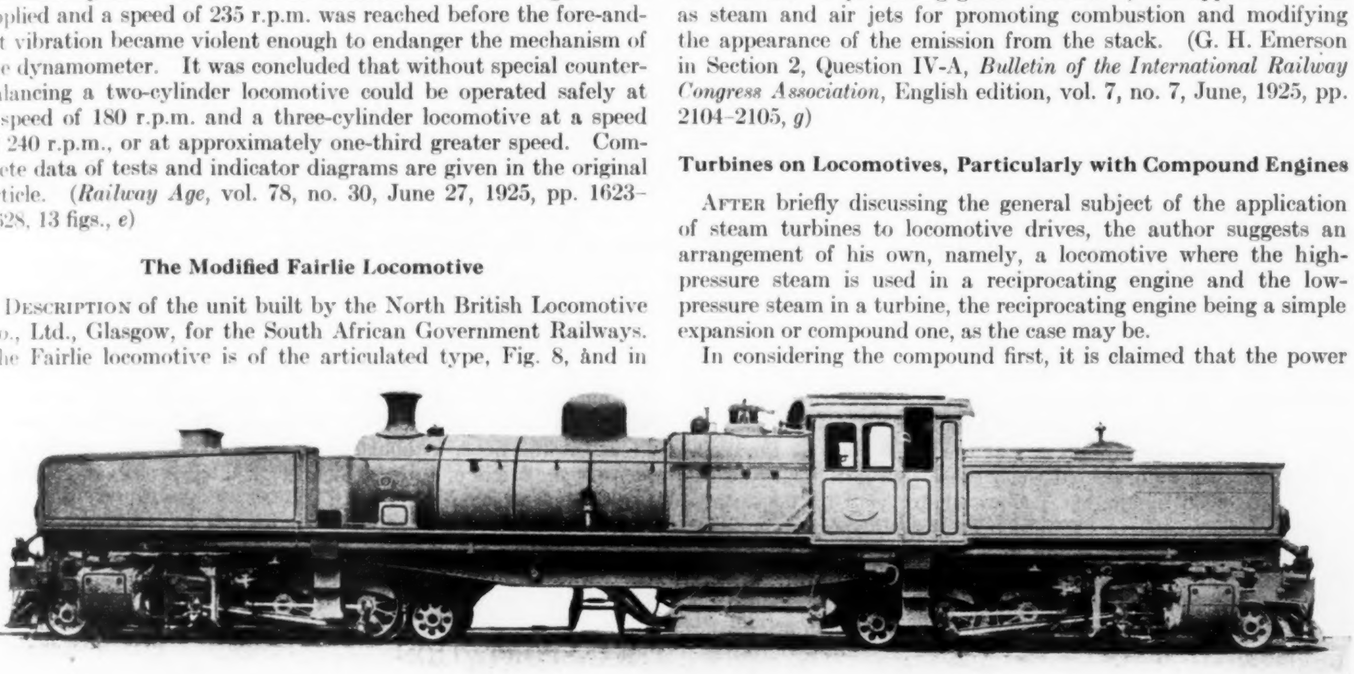


Fig. 8 Modified Fairlie Locomotive

the past consisted of two motor units working as bogies carrying a cradle on which was mounted a double-barreled boiler with single firebox.

In the new design a modified boiler arrangement is used which is claimed to allow of the largest possible boiler within the limits of the loading gage without any sacrifice on flexibility on curves. Instead of the former double-barreled boiler a single-barreled one is employed, water tanks and fuel bunker being carried at either end on rigid extensions of the boiler cradle. Ample room is thus provided for the boiler, while the rigid connection of boiler and tanks obviates the use of flexible tank connections and facilitates the fireman's work in shoveling fuel from the bunker to the firebox.

Boiler, cradle, and tanks are readily detachable, either severally or as one unit. The original article shows the front and rear bogies and tanks detached from the central frame. A recess is provided with the front tank to permit the withdrawal of superheater elements and cleaning of boiler tubes. The boiler is of the Belpaire type and works at a pressure of 180 lb. per sq. in.

The rolling gradient of the South African Railways' new North Coast Line from Durban to Stanger is one in fifty-five, with 528-ft.-radius curves. On this section the new engine is working freight trains of 570 short tons and passenger trains up to 300 tons. (*Mechanical World*, vol. 78, no. 2009, July 3, 1925, p. 3, 3 figs., *d*)

Fuel and Its Combustion on Locomotives

Special report covering North and South America, with particular attention to the United States, made to the International Railway Congress Association.

Among other things it is stated that pulverized fuel has not been found practical for locomotive use, because of limited combustion space available in the locomotive firebox of conventional design. Colloidal fuel is still in the experimental stage.

The mechanical stoker is used on locomotives of greater tractive capacity than about 50,000 lb., which are considered to be of a capacity greater than can be developed consistently by hand firing. Four thousand pounds of coal per hour for limited periods is considered the practical limit for hand firing. There are about 8000 mechanical stokers in use in North America.

The factors in the reduction on the visible products of combustion in the order of relative importance are design and maintenance of the locomotive, character and grade of fuel, operation of the locomotives by engineers and firemen, appliances such as brick arches for promoting good combustion, and appliances such as steam and air jets for promoting combustion and modifying the appearance of the emission from the stack. (G. H. Emerson in Section 2, Question IV-A, *Bulletin of the International Railway Congress Association*, English edition, vol. 7, no. 7, June, 1925, pp. 2104–2105, *g*)

Turbines on Locomotives, Particularly with Compound Engines

After briefly discussing the general subject of the application of steam turbines to locomotive drives, the author suggests an arrangement of his own, namely, a locomotive where the high-pressure steam is used in a reciprocating engine and the low-pressure steam in a turbine, the reciprocating engine being a simple expansion or compound one, as the case may be.

In considering the compound first, it is claimed that the power output of the engine is limited today within a narrow range by the diameter of the low-pressure cylinders. The new arrangement will permit utilizing the low-pressure steam with far greater freedom in as far as considerations of space are concerned. Furthermore, because of the elimination of the low-pressure cylinders the external moving parts will be limited to those only which are actuated by the high-pressure cylinders, which will considerably simplify the transmission mechanism. The low-pressure steam will be utilized better in an engine of the rotary type such as a steam turbine, and the back pressure on the high-pressure cylinders will be more uniform. The turbine is also more efficient than a reciprocating engine on low-pressure steam.

The most important advantage which the author sees in the arrangement advocated by him is that it will be comparatively easy to effect the condensation of a predetermined part of the steam employed. Lately several designs have been offered of turbine-driven locomotives provided with condensers, but to condense an enormous amount of steam still requires an elaborate equipment which it will be hardly possible to maintain in proper condition in actual service. There is also a considerable consumption of power for providing the draft required by the condensers. Most of the tests carried out on turbine locomotives have been made in cold countries where the matter of cooling is comparatively easy to solve, but one may ask what will happen in hot countries. (H. Durin in *Arts et Métiers*, vol. 78, no. 56, May, 1925, pp. 173–175, *g*)

STEAM ENGINEERING

Determination of Viscosity of Water Vapor

Equations for the flow of steam in pipes involve a knowledge of the viscosity of steam. Because of this an investigation for its determination was undertaken in the laboratories of the Technical High School at Karlsruhe. The paper describes the elaborate installation employed and the methods selected, and gives numerical data as to viscosity as found under various conditions.

Among other things, it has been found that the viscosity of water vapor is a function of pressure and temperature. For pressures from 1 to 6 atmos. abs. the viscosity increases in a straight-line

relation to the pressure, while from 6 to 10 atmos. abs. the increase in viscosity is more rapid than the increase in pressure.

According to the kinetic theory of gases, the internal friction of a gas is independent of its density, i.e., of pressure. This requirement, if one may judge from the present tests, has been satisfied only within a certain range of pressures. On one hand it was found that at very low pressures, below atmospheric, the viscosity falls off materially; this has been shown also by previous experimenters (Kundt and Warburg). However, in explanation of this peculiar condition, the assumption has been made that viscosity does not really decrease as much as the experiments would indicate. But the assumption that the gas in contact with the walls has a zero velocity no longer holds under these conditions, and instead, a gliding of the gas takes place at the walls, with the result that the viscosity appears to be less than it actually is. Whether this is so or not has not yet been proved.

The functional relation derived from the kinetic theory of gases to the effect that the viscosity is independent of pressure holds good only within the region of the ideal state of a gas, because it is based on the assumption that the volume of the molecule, which is considered as a perfect elastic ball, is small in comparison with the space available to each molecule of gas for its movements. In other words, the assumption is that the space filled by the molecules of gas will be neglected when considered in reference to the total space filled by the gas. However, for very high pressures, and even for atmospheric pressure in the case of vapors, when taken in proximity to the saturation line, this assumption, because of the much greater density obtaining, does not hold good. Because of this the viscosity of gases at high pressures and the viscosity of vapors at even lower pressures show variations not in accordance with the general theory.

The author gives general expressions for the viscosity of water vapor. (Dr. of Engrg. H. Speyerer, Vienna, in *Forschungsarbeiten auf dem Gebiete des Ingenieuresens*, published by the Verein deutscher Ingenieure, no. 273, 1925, pp. 1–18, 13 figs., *etA*)

TESTING AND MEASUREMENTS

The Degree-Day

With the increasing demand for city gas for house heating and industrial heating purposes it became necessary to have some word or phrase which would enable engineers to make comparisons between heating loads under different climatic conditions or at different points. The "degree-day" which was the result, is the product of a degree of temperature and a time factor of one day.

In studying actual domestic heating conditions, the American Gas Association determined that the minimum temperature of bodily comfort in the home is reached below 65 deg. fahr. In other words, below this point heat is required, as the average daytime temperature in the house drops below 70 deg. fahr.

If, then, we have a mean daily temperature of 60 deg. fahr., it is evident that, for this day, we can measure the heat requirements by the figure "5 deg-days," whereas, had the mean temperature been 55 deg. fahr., the requirement would be measured by "10 deg-days," or would be twice as great. For a week or a month or a heating season, the aggregate heating load of any locality may be expressed in units that permit comparison with that of any other point. Likewise the heating requirements at any given place during the heating season may be compared with those of any other heating season by the "degree-day." So a "mild winter" or a "hard winter" become definable in accurate units that may be translated into B.t.u. Basing these units on the very accurate data of the Weather Bureau a high degree of accuracy results. (*Journal of the Western Society of Engineers*, vol. 30, no. 6, June, 1925, p. 100, *g*)

Automobile-Noise Measurement

The paper here considered is really far broader than its title, as the author takes up the general subject of noise measurement, together with its specific applications to automobile testing. He defines noise as "sound disagreeably out of place," and presents an illustrated and detailed description of the mechanism of human hearing, according to studies made in the interests of telephonic transmission of maximum effectiveness, enumerating and explaining the devices developed for evaluating the sources of sound and its modes of propagation and amplification—because the noise problem as it concerns the automotive engineer must be considered not only as a problem in physics but also as a problem in the physiology of hearing.

An automobile can be considered to be composed of a number of acoustic resonators having varied degrees of coupling between them, and comparisons are made of the velocity of sound propagation through the different materials with that of its transmission in air, the velocity being greater in the structural material. The apparatus used for the detection of noise and its measurement consists of varied types of equipment, divided into two classes; one includes the contact type and the other the air-impact type, both being exemplified and discussed.

Following an enumeration of the different detectors and auxiliary apparatus in use and comments upon the methods employed, it is stated among other conclusions that it seems advisable to base loudness measurements of automobile noise upon the difference of energy between the measured sound and an arbitrary standard of sound which is the threshold of normal hearing; that to locate the origin of automobile noise, it frequently is sufficient merely to detect the noise without measuring its loudness; and that to identify the origin of automobile noise, it often is of value to ascertain its component frequencies.

A bibliography on the subject is appended. (H. Clyde Snook, Bell Telephone Laboratories, Inc., New York City, in *Journal of the Society of Automotive Engineers*, vol. 17, no. 1, July, 1925, pp. 115–124, 17 figs., *eA*)

TRANSPORTATION

Street Congestion and Motor-Sales Saturation Point

Hitherto studies regarding the saturation point for motor-vehicle consumption in this country have been based largely upon estimates of the number of people who possess money enough to buy a motor vehicle; the speaker believes, however, it more probable that some approach to a true estimation of a saturation point will be obtained from a proper estimate of the available road space in the country upon which motor vehicles can be operated. Citing production statistics of motor vehicles for 1923, 3,629,237 passenger cars and 392,760 trucks, he estimated that, allowing a 12-ft. length for a passenger car and a 20-ft. length for a truck, with two vehicle lengths between vehicles, they would represent a single line of traffic 34,644 miles in length or a moving line of cars and trucks, 10 vehicles abreast, that would extend from New York City to San Francisco. Further, he stated that motor-vehicle production for one year had placed upon the roads a greater mileage of single-line traffic than there were miles of Federal Aid highways constructed from 1917 to 1925.

After showing lantern slides of street congestion in Chicago, the speaker mentioned the motor coach, especially the double-deck type, as an exceptionally efficient vehicle for moving passengers in congested areas, and stated that it is only within the past several years that city-planning engineers have given the motor coach much attention, seeming also to have ignored entirely the individually owned and operated passenger car as a transportation medium. The time is not far distant when certain sections of very large cities will be closed entirely to the privately owned and operated automobile, a likely substitute therefor being the taxicab. (Address by A. W. S. Herrington, Chief Engineer of the Engineering and Design Section of the Motor Transportation Division, Quartermaster Corps. before the Washington Section of the Society of Automotive Engineers, June 11, 1925. Abstracted through the *Journal of the Society of Automotive Engineers*, vol. 17, no. 1, July, 1925, pp. 125–126, *g*)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Test Code for Steam Turbines

Tentative Draft of a Code in the Series of Nineteen being Formulated by the A.S.M.E. Committee on Power Test Codes

THE Main Committee on Power Test Codes takes pleasure in presenting to the members of the Society the Test Code for Steam Turbines for criticism and comment. The Individual Committee which developed this draft of the code consists of Messrs. C. H. Berry, Chairman, I. E. Moultrop, Secretary, E. H. Brown, M. D. Church, H. Dahlstrand, F. Hodgkinson, W. J. A. London, S. A. Moss, C. C. Thomas. It is believed that in its present form this code meets the needs of all groups which from time to time have a part in the making of acceptance tests of this type of apparatus.

In 1918 the Committee on Power Test Codes was organized by the Council of the A.S.M.E. to revise and enlarge the Power Test Codes of the Society published in 1915. This Committee consists of a Main Committee of twenty-five members under the chairmanship of Fred R. Low and nineteen individual committees of specialists who are drafting test codes for the various prime movers and the other auxiliary apparatus which constitute power-plant equipment.

The Individual Committee, the Main Committee, and the Society will welcome suggestions for corrections or additions to this draft from those who are especially interested in the manufacture, use, and testing of steam turbines. These comments should be addressed to the Chairman of the Committee in care of The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y.

Introduction

1 General instructions for the conduct of tests are given in the division of this code entitled "General Instructions," and that division should be studied carefully and followed in detail. In particular, before any test is begun, the object sought must be determined and agreed upon by all interested parties, and must be borne in mind throughout the test.

2 The "Code on Definitions and Values," defines certain technical terms and numerical constants, which are used throughout this code with the meanings and values there established.

3 The "Code on Instruments and Apparatus," discusses fully the selection, calibration, application, and use of instruments for test observations. Those parts of that code covering instruments to be used in a turbine test should be studied in detail, and followed accurately and fully in arranging for and conducting tests. In the following paragraphs specific references are given to paragraphs in the "Code on Instruments and Apparatus."

4 This Code refers specifically to the testing of steam turbines, and comprises instructions peculiar thereto. It establishes rules for tests to determine

a Steam Rate (For checking the thermal economy characteristics of the turbine.)

b Capacity (For checking turbine capacity under various circumstances, as, for example, the maximum attainable capacity with certain steam conditions, etc.)

5 From the standpoint of steam utilization, steam turbines may be classified as follows:

a Simple turbines (no extraction, no reheating, no mixed-pressure features—all steam enters at one pressure, and all leaves at one pressure)

b Turbines with intermediate reheating of the steam

c Regenerative turbines, from which steam is extracted only for heating feedwater, and for which the performance is to be determined on a thermal basis, the turbine being credited with the heat returned to feedwater by steam extracted from the turbine

d Extraction turbines, from which steam of unknown quality is extracted for exterior purposes

e Mixed-pressure turbines

f Combined mixed-pressure and extraction turbines.

6 For the present, this code provides instructions for steam-rate tests for cases *a*, *b*, and *c* only, with the essential auxiliaries, such as oil pumps, generator fans, etc., whose operation is strictly necessary to the operation of the turbine itself.

7 The determination of steam rate in cases *d*, *e*, and *f* is a very difficult matter, wherefore it is deemed neither feasible nor desirable to reduce such testing to the basis of a general code in the present state of the art. The principal difficulty lies in the determination of the quantity or quality of low-pressure steam, either or both of which are required in each of these cases. Such determinations are in the nature of research rather than of testing at the present time. In specific cases conditions may be such that this code may be followed, as, for example, if the low-pressure steam is superheated, or if the guarantee or other basis of the test is so drawn that the code may be followed. An instance of this latter case is the guaranteeing of extraction or mixed-pressure turbines on the basis of their steam rate without extraction or without the admission of low-pressure steam.

8 This code also provides instructions for measuring the load on an electric generator driven by a turbine. In the case of other special applications, those codes are to be consulted which treat of testing the various types of driven units, and their instructions for the measurement of load are to be followed.

Items on which Agreement must be Reached

9 Throughout this code there are necessarily presented alternative methods for various details of conducting the test and developing the results, and various questions affecting the results. It is necessary that all parties to a test consider these matters before the test begins, and agree, preferably in writing, concerning the disposition of each item. In some instances, conditions will determine the question; in others, the consideration of convenience, time available, expense, and the like. In order that none of these items may be overlooked, they are all cited below.

Par. 12 Rejection of test runs
Par. 13 "Such other items"
Par. 16ff Instruments
Par. 18ff Accuracy of load observations
Par. 23 Method of determining total steam
Par. 26 Method of determining condenser leakage. See "Code on Instruments and Apparatus."
Par. 28 Arrangements for checking water level in the condenser hotwell or inlet pipe to the condensate pump.
Par. 29 Arrangements for dealing with water used for sealing, etc.
Par. 30 Vapor passing through air pump[1]
Par. 31 Sealing steam
Par. 36 Apparatus between the throttle and the turbine
Par. 38 Wet steam
Par. 41 Type of exhaust pressure gages and location and number of vacuum-gage pressure connections
Par. 64 Shall the preliminary test be made official? This can be decided only after the preliminary test has been finished
Par. 67ff Method of holding constant load
Par. 70 Overload devices
Par. 72 Position of hand valves
Par. 74f Duration of runs
Par. 81 Measurement of initial steam pressure
Par. 84 Method of determining corrections for variations in operating conditions.

Results of Test

10 In a steam-rate test the object will be attained by the statement of results in terms of the quantities

a Steam rate

b Heat consumption or its converse, thermal efficiency

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[1] Here and elsewhere in this code, the term "air pump" is used to designate equipment for the removal of uncondensable gas from the condenser, whatever the type of such equipment may be.

c Engine efficiency (Rankine). (See "Code on Definitions and Values.")

These results must be accompanied by the values of other quantities necessary for their proper interpretation, such as steam pressures and temperatures, water temperatures, etc.

11 The results of a capacity test will take various forms, depending upon the nature of the guarantee. Usually, however, the results will take the form of a statement of the load developed and of the concomitant steam and exhaust conditions, governor control positions, valve adjustments, etc.

Rejection of Test Runs

12 If serious inconsistencies arise, either during a test run (see Par. 77), or during the computation of results from a series of runs, the run or runs shall be rejected, in whole or in part, and repeated, if this is necessary to secure the objects of the test.

Essential Data

13 Under this heading are included the observations of load, total steam, condenser leakage, steam conditions at throttle and exhaust, speed, time, and such other items as may be called for in each specific case.

14 *Instruments and Accuracy.* For the observation of these quantities accurate instruments shall be used, and these shall be selected, calibrated, applied, read, and corrected as specified in the relevant sections of the "Code on Instruments and Apparatus." In the following paragraphs, the limits of error are mentioned in some cases. In every case these indicate a maximum error within which observations may reasonably be expected to fall. If these limits are exceeded, the results will be correspondingly unreliable.

15 Instruments which are liable to failure or breakage in service, such as pressure gages and thermometers, shall be duplicated by reserve instruments, properly calibrated, which may be put into service without delay.

16 *Necessary Instruments.* The following instruments are required for a steam-rate test. They are discussed in detail in succeeding paragraphs:

For a turbine running alone, a dynamometer of a type suitable to the turbine and the circumstances of the test (Par. 18)

For a turbo-generator, instruments for the measurement of generator output, as called for in Instruments and Apparatus, Par. .....

For a turbine exhausting to a surface condenser, water-weighing tanks and scales, or properly constructed volumetric measuring tanks, and suitable instruments for the measurement of condenser leakage (Par. 23)

For a turbine exhausting to atmosphere or to a jet condenser, water-weighing or measuring tanks and a boiler suitably isolated, or facilities for calibrating the first-stage nozzle block (Par. 23)

A Bourdon or dead-weight gage for measuring steam pressure just ahead of the throttle (Pars. 36 and 37)

If saturated steam is used, suitable instruments for determining its quality at the throttle (Pars. 36 and 38)

If superheated steam is used, proper thermometer and well for determining its temperature just ahead of the throttle (Pars. 36 and 39)

Mercury columns, dead-weight gages, or accurate Bourdon gages, for determining vacuum or pressure of the exhaust steam (Pars. 41 and 49)

Barometer (Par. 41)

Thermometers for determining temperature of mercury columns and barometer

Speed indicator (Par. 51)

Clock, or synchronized watches (Par. 53).

17 For a capacity test many of the same instruments are required as for steam-rate test. Of course there need be no provision for determining the total steam. In addition to such instruments as have been noted above, it may be necessary to provide indicating devices, scales, etc. whereby the time of opening and the extent of opening of various turbine-control valves can be observed.

18 *Load on Turbine.* The *load* on the turbine is the power being developed expressed in brake horsepower or in net kilowatts delivered (See "Code on Definitions and Values," items 19, 20, and 114). In some cases periodic observations of this quantity by indicating instruments serve as the basis for computation of the output, and in such cases the load (or data from which the load may be computed) must be observed at frequent intervals (not ordinarily more frequent than every 5 minutes), and with a high degree of accuracy. In other cases the integrated output is measured independently of the load, and the average load is calculated from the measurement of integrated output and time.

19 In either case it is necessary to observe the momentary load (or data from which the load may be computed) at intervals sufficiently short to give a true indication of the nature of any variations which may occur. In some cases this may require frequent, in other cases rare, observations. Such data are of value both as a guide to operation and because variations affect the economy of the turbine. For this reason the momentary load shall never be more than 5 per cent different from the average load. For the purpose of general control, when the output is measured by accurate integrating instruments, a low degree of absolute accuracy of indicating instruments is sufficient; variations, on the contrary, must be accurately noted. A switchboard electrical instrument may be used without calibration. In some cases the rate of steam flow as indicated by a flowmeter, the steam pressure in intermediate stages of a turbine, or the torque exerted by a constant-speed turbine as indicated by a dynamometer, may be as significant in indicating load variations as the readings of an indicating wattmeter connected to a turbo-generator.

20 The load on a turbine running alone may be measured by means of any suitable dynamometer, such as friction brake, torsion meter, or the like. See "Code on Instruments and Apparatus."

21 The load on a turbo-generator is the net output of the generator, which shall be determined as directed in Instruments and Apparatus, Par. .....

22 In the case of other special turbine applications, those codes are to be consulted which treat of testing the various types of driven units, and their instructions for the measurement of load are to be followed.

23 *Total Steam.* The total steam shall be determined as follows:

a For a turbine exhausting to a surface condenser:
By weighing or measuring the condensate.

b For a turbine exhausting to atmosphere, or to a jet condenser:
1 By weighing or measuring the water fed to a boiler whose only open outlet is the pipe leading to the turbine under test
2 By removing and measuring flow through the first-stage nozzle block.

24 When water is weighed, scales shall be fitted with tanks arranged so that a continuous flow can be cared for, and the scales shall not show an error greater than 2 in 1000. These shall be arranged, operated, and calibrated as provided in the "Code on Instruments and Apparatus."

25 When water is measured, the volumetric tanks used shall be of the type described in the "Code on Instruments and Apparatus." They shall be accurately calibrated by actual weighing, and a correction shall be determined and applied for water temperature.

26 In connection with any test in which the total steam is determined by weighing or measuring the condensate from a surface condenser, the condenser must be tested for leakage of cooling water into condensate as prescribed in the "Code on Instruments and Apparatus." If this leakage exceeds the limits given below, the condenser is not in condition suitable for a reliable test, and it shall be made satisfactorily tight before the test. These limits depend upon the rated capacity of the turbine, and are expressed as a percentage of the rated, or guaranteed, steam flow at rated load.

Less than 500 kw., or 700 hp.	1.00 per cent
500 to 1000 kw., or 700 to 1400 hp.	0.75 per cent
Over 1000 kw., or 1400 hp.	0.50 per cent

27 If condenser leakage is determined by silver nitrate titration (for salt water), or by the electrolytic-conductance method, observations shall be made every half-hour during the turbine steam rate test, in addition to the observations made during the preliminary leakage test. For these methods, see the "Code on Instruments and Apparatus." If the observations made during the steam-rate test indicate an increase of condenser leakage above the allowable limit, the steam-rate test shall be discontinued, and the condenser shall be made satisfactorily tight before resuming the turbine test.

28 The water level in the condenser hotwell or inlet pipe to the condensate pump shall be checked at the beginning and end of a run, and at intermediate points which mark the division of the run

into periods, for each of which the steam rate is to be determined; and correction shall be made for any outstanding change in the quantity of water stored therein. It is highly desirable that means be employed for holding the condensate level constant throughout the test, as for example, by throttling the discharge of the condensate pump. One satisfactory way of obtaining the desired result is to note that a gage glass on the inlet pipe to the condensate pump shows a constant level. If the volume of the pipe between the gage glass and the condensate pump is small, it will be sufficient to see that the gage glass is always empty. The resulting error can easily be computed, and will usually be negligible. It shall be determined that there are no other points at which water can accumulate. If such places cannot be avoided, proper correction shall be made for each of them just as for the hotwell. All lines leading from and to the condenser, other than the air and condensate pump suctions and turbine drips, shall be blanked off, or fitted with two valves with a bleeder between them, or, if this is impossible, means shall be provided for determining the quantity of water passing through each such line.

29 If condensate is used for any purpose, such as for sealing pump glands, turbine glands, the atmospheric relief valve, etc.

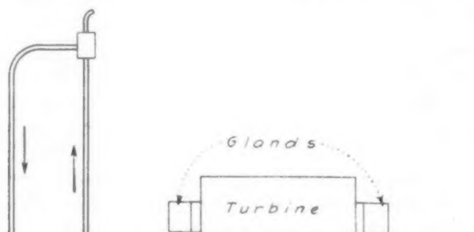

Fig. 1 Diagram of Piping for Supplying Condensate to Shaft Glands

the quantity so used must be determined and included in the total. Condensate lost through leaky valves must be determined and included, if it has not previously been accounted for. Water other then condensate entering the condenser, or other parts of the system, and weighed or measured condensate reëntering the condenser, must be determined and deducted from the total. It is plainly desirable that the arrangement of piping, etc. be such that the need of such corrections to the test results be reduced to a minimum. For instance, in the case of turbines equipped with water glands with condensate employed for sealing, they may be piped up as shown in Fig. 1. Water is taken from the line between the condensate pump and weighing tanks and pumped to the desired head for sealing the glands, and the overflow is returned to the same pipe. Of course, any external leakage from such glands must be collected, properly weighed, and charged against the turbine. Similarly, any gland or water seal employed to preclude air leakage to parts connected with the condenser and under vacuum, may be piped up in the same manner and cognizance taken of leakage outward only in the computing of the test results.

30 The steam removed with air by the air pump is usually a negligible quantity, and no attempt to determine it or to correct for it is necessary or justified, except in unusual cases, as, for example, when an air pump of exceptionally large capacity is installed. If such is the case, some method of correction must be agreed upon before the test begins.

31 With suitable precautions, the *weighing or measuring of boiler feedwater* should give results accurate to within plus or minus 3 per cent. To secure even this degree of accuracy, it is necessary to reduce the effect of the inevitable uncertainties of this method by the exercise of the greatest care, and by the prolongation of each constant-load run to not less than ten hours duration. See "Code on Instruments and Apparatus" for detailed instructions for the conduct of tests by this method.

32 In cases in which it is impossible to arrange for weighing or measuring condensate or boiler feedwater, the steam rate of a turbine may be determined by *removing the first-stage nozzle block* and determining the relation between total flow and initial steam pressure. This method shall not be used in cases in which the absolute exit pressure of the first-stage nozzles exceeds 58 per cent of the absolute initial steam pressure. Several precautions are necessary. The nozzle block must be replaced in such a way that it is certain that there is no leakage of steam around it. For this reason it is recommended that this method never be used if it is necessary to break a high-pressure steam joint within the turbine in order to remove the nozzle block. The determination of steam flow through the nozzles must be made by the actual full flow of steam into a surface condenser, the condensate being weighed or measured. If the first-stage nozzles are in two or more groups, these may of course be tested separately. Care must be taken in measuring steam pressures during both the nozzle test and the turbine test, the pressure connections being located at precisely the same points in both tests. The initial pressure must be taken in the chamber from which the nozzles draw steam directly—this is sometimes called the "bowl" pressure. The steam pressure at the exit from the nozzles must not exceed 58 per cent of the steam pressure in the chamber from which these nozzles receive steam. The chamber into which the discharge occurs and which connects the nozzle block to the surface condenser must be of ample size, so as to avoid cross-currents or eddies, and so as to give a uniform static pressure throughout the region into which the nozzles discharge. The static pressure must be taken at one or more points in this chamber to be certain that the nozzle pressure therein does not exceed 58 per cent of the initial absolute pressure.

33 Incidental steam supplied to any auxiliary which is included in the unit being tested, but not included in the main condensate, can usually be determined by a steam meter. In this connection the consideration of relative accuracy is of special importance. For example, suppose a turbine to be served by auxiliaries whose steam consumption is 1 per cent of the steam consumption of the main unit. For every 1000 pounds of steam used by the main unit, the auxiliary device receives 10, and the two together receive 1010. The main supply shall be measured with great care—assume that it has been measured with a probable error of not more than 1 in 1000. This will be an absolute error of 1 in 1010 of the aggregate steam consumption of the turbine and its auxiliaries. A meter measuring the smaller quantity of steam passing to the auxiliaries need not have a greater absolute accuracy than that attained in measuring the larger quantity, that is, it may have an absolute error of 1 in 1010 of the aggregate steam consumption. Since this meter measures only 10 in 1010 of the aggregate steam consumption, its error relative to its own reading may be as great as 1 in 10 without introducing errors in the final result greater than those probable from other sources. Accordingly, relatively small portions of the total steam may be measured by flowmeters carefully adjusted and accepted with only approximate calibration, or, in extreme cases, without calibration. On the other hand, the performance of auxiliary devices is not without interest, and the steam supplied to them should always be measured with as great accuracy as the circumstances permit. The fact that large relative errors are permissible in such quantities is no justification for failing to take ordinary precautions to minimize all known and controllable errors.

34 The steam escaping from all steam seals shall be eliminated or, if it cannot be reduced to a quantity which is agreed to be negligible, it shall be caught, condensed, and weighed or measured. The steam may be condensed either by an air-cooled or water-cooled surface condenser or in a bucket of ice water, weighed before and after.

35 In determining the total steam passing the throttle of a turbine operating on the *regenerative cycle*, several precautions are necessary. There are several ways of disposing of the drips from extraction heaters, but all methods finally discharge the drips into

the main stream of water leaving the unit to return to the boiler room. Care must be taken to intercept the water stream, for weighing or measuring, beyond the point at which such drips return, so that the condensate from all steam which passes the throttle, shall be included. It will often be the case that the water at this point is too hot to be weighed or measured accurately, because of excessive evaporation. In such cases it will be necessary to install temporarily a heat interchanger for cooling the water to 140 deg. fahr. or below. The detailed study of the performance of regenerative turbines, involving determination of individual quantities of steam extracted, is beyond the scope of this code.

36 *Condition of Steam Supplied.* The initial condition of steam supplied must be determined in the steam line on the boiler side of the throttle valve[1] and strainer and as close as possible to the throttle valve, except that if apparatus which will affect the turbine performance, and which is not contemplated in the efficiency guarantee is interposed between the throttle valve and the turbine, the initial steam condition must be determined between such apparatus and the turbine, or such apparatus must be removed for the conduct of the turbine test.

37 The *pressure* shall be measured by means of a calibrated gage of the Bourdon or dead-weight type, whose instrumental error must not exceed one per cent of the pressure indicated. Reference shall be had to the "Code on Instruments and Apparatus," and all the precautions there given shall be observed. The gage connection shall preferably be in a straight run of pipe at least 5 diameters long, and $2^1/_2$ diameters from the discharge end of the pipe.

38 The *quality* of wet steam shall be determined by means of a [type to be specified] calorimeter, as prescribed in "Instruments and Apparatus." It must be borne in mind that the moisture content of wet steam has a disproportionately large effect on the turbine steam rate, and that the quality of steam is very difficult of accurate determination. Accordingly, when a turbine under test is supplied with wet steam, the test results are very likely to be of considerably lower accuracy than those obtained with superheated steam. If the calorimeter indicates moisture in excess of 2 per cent, the steam-rate test cannot be regarded as reliable. Every effort should be made to avoid testing any turbine with a wet-steam supply. Even in the case of turbines designed to run on saturated steam, it is better to test them with steam somewhat superheated, say 25 deg. fahr., and then, by means of suitable correction factors, to make whatever comparisons are called for.

39 The *temperature* of superheated steam shall be measured by means of a mercury-in-glass thermometer, having an engraved stem at least 12 in. long, accurate to 1 deg. fahr., which will not break its thread on cooling, and which will return to within one degree of its original reading after quick heating to steam temperature and cooling to room temperature. This thermometer shall be calibrated both before and after the test by one of the methods prescribed in the "Code on Instruments and Apparatus." Should a thermometer be broken during a test, it shall be replaced by a similar instrument, likewise calibrated, and such reserve thermometer shall be provided and calibrated before the beginning of the test. The steam thermometer shall be inserted in a finned well of the type described in the "Code on Instruments and Apparatus," filled with mercury, fused sodium nitrate, soft solder, or other material. The well shall preferably be located in the middle of a straight run of pipe five diameters long. Stem corrections shall be made as provided in the "Code on Instruments and Apparatus," temperature of emergent stem being taken by means of an auxiliary thermometer.

40 *Intermediate Reheating.* If the steam is withdrawn from the turbine for reheating, it will be necessary to observe the pressure and temperature of the steam at the point of withdrawal and at the point of return to the turbine, after reheating. All of the instructions given in Pars. 37 to 39 apply to these observations also.

41 *Condition of Steam Rejected.* For consistent accuracy, *exhaust pressure* must be measured much more accurately than initial pressure.

42 For condensing turbines, exhaust pressure shall be measured by means of carefully constructed mercury columns, accurate to 0.01 in. of mercury, used with all precautions and corrected for all observable causes of error. See "Code on Instruments and Apparatus," Pars. ..... Each column shall be mounted within not more than six feet of the pressure hole in the conduit or casing, and shall be connected to the exhaust passage as provided in the "Code on Instruments and Apparatus." Each gage must be so mounted as to be as nearly as possible free from vibration.

43 If the conduit connecting the turbine exhaust nozzle and the condenser inlet nozzle is approximately straight, the exhaust pressure shall be measured as far as possible from the turbine exhaust flange (toward the condenser), but at a distance not to exceed three diameters of a circle of area equal to that of the exhaust conduit. Each vacuum gage shall be connected to a static pressure hole normal to the inner wall of the conduit, and not smaller than $^1/_2$ in. standard iron-pipe size. There shall be approximately one such hole and gage for each 16 sq. ft. of exhaust-conduit area, but in no ordinary case less than 2 nor more than 6 sq. ft. The pressure holes shall be distributed as uniformly as possible around the periphery of the conduit. When both turbine and condenser are tested together, in many cases common readings of exhaust pressure will suffice for both tests.

44 If there is no straight conduit between turbine and condenser, measurement shall be made in exactly the same way, but the pressure holes shall be located within 12 in. of the face of the turbine exhaust flange, and at that plane where the steam flow is most likely to be smooth. If both turbine and condenser are tested together, two sets of readings will in general be required.

45 If the vacuum gages show serious discrepancies, the situation must be specially studied. Possibly more gages may be necessary.

46 The readings of all gages, corrected to inches of mercury at 32 deg. fahr., shall be averaged, and subtracted from a similarly corrected reading of a mercurial barometer, to give the absolute exhaust pressure. Correction must be made for the difference in elevation of the barometer and vacuum gages, if any. See "Code on Instruments and Apparatus," Pars. ..... The temperature correction may be made negligibly small by maintaining vacuum gages and barometer at approximately the same temperature, but if this is done, the temperature correction shall be computed for sufficient cases to demonstrate that it is negligible.

47 For non-condensing turbines, the exhaust pressure shall be determined by means of a mercury column for pressures below 20 lb. per sq. in. gage, or by Bourdon gage, or dead-weight gage, for higher pressures, as prescribed in the "Code on Instruments and Apparatus," connected to a hole flush with the interior passage, within 12 in. of the plane of the face of the turbine exhaust flange. The gage reading will be added to or subtracted from the barometer, depending upon whether the exhaust pressure is above or below atmospheric, exactly as described above for condensing turbines.

48 Exhaust pressure shall never be computed from temperature observations.

49 When a turbine is set up for test on a temporary base there shall be connected to the exhaust opening an exhaust conduit of the full size for which the turbine is arranged. This conduit beyond the exhaust opening shall preferably be straight for about 18 in. If it is not possible to arrange straight pipe of this length the exhaust conduit shall be arranged in exactly the same way as will occur in the permanent installation. These stipulations are made because of the inaccuracy in measurements of exhaust pressure with small-size exhaust pipes or other makeshifts for the temporary test set up.

50 *Temperature of Condensate.* For turbines operating on the regenerative cycle, it is important to observe accurately the final temperature of water leaving the turbine unit to return to the boiler room. Water temperatures at intermediate points in the heating process are of interest, but are not involved in any calculations called for by this code.

51 *Speed.* The *speed* should be observed at intervals sufficiently short to give a true indication of the nature of any variations which may appear. When the output of the turbine is determined by a brake or similar dynamometer, speed must be observed with the utmost accuracy, or else the total revolutions must be counted by some suitable device. See "Code on Instruments and Apparatus."

[1] By throttle valve is meant the steam stop valve used to start or stop the turbine, which, when open, gives a practically negligible pressure drop. This should be distinguished from governor or regulator valves. Of course the steam strainer must be clean, and cause no abnormal drop in pressure.

52 For tests of speed regulation, see the code for testing speed-responsive governors.

53 *Time.* The exact *time* (accurate to one second) shall be recorded for starting and stopping the weighing or measuring of water for each run, and for the initial and final readings of integrating load instruments for each run. If a run is to be divided into parts for checking, the exact time of the intermediate water check and integrating meter reading shall also be recorded. The approximate time (to the nearest minute) of all other observations shall be recorded. Simultaneous observations may be controlled by the use of whistles, bells, or the like, operated from a common electric circuit, with warning and final signals for each reading. Another convenient method is the use of two or more synchronized stop watches. These are started simultaneously (a little practice makes this easy) a few minutes before the beginning of a test run, and are given to the observers of load and water weight, with the understanding that the run will start on the *n*th minute and stop on the *m*th minute, with other intermediate points, if desired. The water weighers should record the time, accurate to one second, on each occasion when the flow of water is diverted from one tank to another. Such data are of great value in checking for errors in weighing, such as the loss of a tank of water.

54 *Other Items.* In certain cases the guarantee may include specific features of turbine operation not covered above. In such cases observation of these features is of course essential. In particular, capacity guarantees often specify that certain valves shall be wide open, or just about to open, or the like; and provision must be made to observe their opening and the turbine load at that time. The details of such provisions depend to so great an extent upon the mechanism of the valves that nothing can be specified in detail in this code.

Secondary Data

55 Under this heading are included quantities whose values are not essential to the computation of the steam rate. They may, however, throw much light on the performance of the turbine, or facilitate the conduct of the test. In some cases, the determination of some of these quantities may be specifically required by the object of the test. In other cases their observation is optional.

56 *Instruments and Accuracy.* Such quantities as are required by the object of the test must be determined to a degree of accuracy sufficient to meet the requirements, and suitable instruments must be chosen to this end. Quantities whose observation is optional may usually be determined with ample accuracy by operating instruments installed with the turbine, if there are such.

57 *Operating Instruments.* All operating instruments installed with the unit may be read regularly if desired. Such of these as are duplicated by test instruments will thereby be calibrated more or less satisfactorily. In any case, the indications of such instruments indicate operating conditions in a manner comparable with regular operation. When such instruments are read, the observations shall be recorded on separate log sheets clearly marked. No computations of results may be based on such observations.

58 *Stage Pressures.* In a test of a multistage turbine the observation of steam pressure in one or more of the stages gives valuable data on the behavior of the machine by indicating variations in steam flow, frequently bringing to light inconsistencies in other observations, and such observations shall be made whenever it is possible to connect the requisite gages. Such observations are of special importance for turbines operating on the regenerative cycle.

59 *Temperatures.* It is usually helpful in interpreting test results to have observations of certain temperatures. When saturated steam is supplied, its temperature serves as a check on the indications of the pressure gage; however, it must be borne in mind that steam may be slightly superheated by certain types of boilers, even though no superheater is installed. In like manner, the temperature of the exhaust steam is a rough check on the readings of exhaust pressure. A thermometer for this purpose, however, must be installed with considerable care and good judgment, lest it indicate the temperature of the metal of the exhaust nozzle rather than the actual steam temperature. In the testing of small turbines running on superheated steam, the exhaust is usually highly superheated, and its temperature is often an important quantity which should be measured with every effort to secure an accurate result.

Preparation for Test

60 Reference should be had to the "Code on General Instructions," and the directions there given shall be carried out in full detail.

61 The calibration of instruments shall be conducted as specified in the "Code on Instruments and Apparatus." If the total steam is to be determined by weighing or measuring condensate, the condenser shall be tested for leakage before the test. See Par. 26, and the "Code on Instruments and Apparatus."

62 The turbine shaft seals shall be tested for leakage of steam out or air in, and shall be adjusted to their normal condition before beginning a test.

63 If the total steam is to be determined by weighing or measuring boiler feedwater, the requirements of the "Code on Instruments and Apparatus," must be carried out before the beginning of the test.

64 A preliminary steam rate, or capacity test, shall be run for the purpose of determining whether the turbine is in condition suitable for the conduct of a formal test, for the checking of all instruments, and for the training of personnel. After this test the manufacturer shall be given an opportunity to adjust the turbine should its performance be in any way unsatisfactory to him or to the purchaser. By mutual agreement, this preliminary test may subsequently be declared formal, and in all respects equally acceptable with subsequent tests. For this reason, the preliminary test should be conducted with all instruments, methods, observers, etc., exactly as though it were known beforehand to be a formal test.

65 For both preliminary and formal tests the turbine must be in commercial operating condition, with the exception of the provisions of Pars. 36, 37(*c*) and 70, and of any specific agreements. That is, when the test is finished, no adjustments shall be necessary prior to putting the machine into indefinite commercial operation. The machine may continue to run after the test, without intermission, if conditions call for such operation, with the exceptions noted above.

Conduct of Test

66 *Constancy of Conditions.* During each run of a test, any influence whose variation may affect the results of the test shall be allowed to become as nearly constant as possible before the run begins, and must be so maintained throughout the run. The use of graphic logs is sometimes of value in determining the proper starting point for a run.

67 In particular, if load variations occur, accurate results cannot be expected. Therefore the turbine load must be maintained at a practically fixed value throughout each run, as provided in Par. 19. This result may be reached by various methods, as follows, when the turbine is running in parallel with other machines.

a Hand regulation of the speed-control mechanism of the governor by a thoroughly skilled operator is often satisfactory with large machines and may be used if all parties agree. In case of small machines load variations are apt to be an appreciable fraction of the output, and it is not possible for an operator to follow them quickly enough to make this method advisable.

b Set the governor for a speed slightly higher than that at which the turbine is to run, and control the steam supply by the manual throttle valve. This method can be used with throttling governors, or with multiple valves when certain valves are put out of control of the governor as in Par. 70.

c Block the governor in such a way as to limit its travel in the opening direction only and adjust the governor for a higher speed than that at which the turbine is to run. The force of the governor will then tend to open the valves throughout the test, the extent of their opening being determined by the point at which the governor is blocked.

68 When a turbine is operated with normal governor control, there is at all times a moderate and continuous variation in the output. It will be noted that methods *b* and *c*, above, eliminate this variation, while method *a* fails to do so.

69 The turbine governor must never be disconnected nor changed nor adjusted in any way which will interfere with its

ability to close all steam valves at any time in case of increased speed.

70 Turbines are often equipped with automatic valves designed to give overload capacity or to give rated capacity when operating conditions (steam pressure, etc.) are poorer than those for which the turbine was designed. If such a turbine is to be run at a load just below that at which these valves are designed to open, it may prove impossible to hold the load sufficiently constant to prevent these valves from popping open from time to time. In such cases it is permissible, and may be necessary, to block such valves shut, or to disconnect their operating mechanism, or to render them inoperative through governor control. Such valves may not be removed or blanked off. If such valves are to be rendered inoperative, there should be a previous agreement among all parties concerned, recognizing the necessity for such action, and prescribing the method to be used.

71 For a turbine running alone, the maintenance of constant load is a special problem in each case. A water rheostat, or special adjustments of load, may be adopted.

72 In tests of throttle-governed turbines in which manually operated valves are provided to nozzle sections, by means of which better steam consumption may be secured at fractional loads, such valves shall remain open throughout the tests, the turbine thus being able to carry the maximum specified load. This requirement is made because the turbine is not in operating condition if, during the test, it is incapable of swinging up to maximum load at any time. On the other hand, manual valves provided for carrying overloads should be closed during tests under operating conditions. Subsequent tests may be conducted with various valves closed, provided the contract, or other previous written agreement, so specifies. See Pars. 97 and 98, regarding the proper selection of critical load values.

73 During a capacity test, unless otherwise specified, the governor must remain as near the limit of its travel as is possible, consistent with continuous governor control. For a turbine running alone, the speed shall be as near the rated speed as this requirement permits.

74 *Duration.* Each constant-load run comprised in a steam-rate test shall continue for not less than one hour when condensate is weighed, or measured. When boiler feed is weighed or measured, each constant load run shall continue for not less than 10 hours. Each test run shall be preceded by a preliminary period adequate to establishing constant conditions, as shown by suitable quantitative observations. See Par. 77. During this preliminary run the load shall be maintained constant and at the same value as during the period during which observations are made.

75 In a capacity test, the load which is to check the guaranteed capacity, together with the nearest attainable approximation to the guarantee operating conditions shall be held substantially constant for not less than five minutes, unless the guarantee contract expressly provides otherwise.

76 *Starting and Stopping.* A test shall start (and stop) with the simultaneous initiation (or discontinuance) of those power or output readings which are read from integrating instruments, and of water weighing or measuring. It is essential that the observations of these two quantities be started and stopped strictly simultaneously. All other readings shall start somewhat before and stop somewhat after these times.

77 During the period preliminary to a test run, readings shall begin after the load and other operating conditions have been held practically constant for about 15 minutes. Steam rate shall be computed for successive periods of from 3 to 15 minutes (such periods need not be of equal duration), and the formal run shall be deemed to have started with the first two such successive periods for which the uncorrected steam rate checks within 3 per cent. This checking of successive periods should be continued throughout the run, as an indication of the constancy of operating conditions.

78 If the output is measured by an integrating instrument, its initial and final readings mark the start and finish of the run with respect to this reading. If, on the other hand, the output is to be computed from a power-indicating instrument, the start and finish will each be noted by the simple observation of the exact time, or the reading of a continuous revolution counter.

79 *Operation at Standard Conditions.* Every effort must be made to run the tests with the rated, or "guarantee," or other standard values of all conditions, such as pressure, superheat, or vacuum. Previous to the test there should be ready the initial calibrations of all instruments involved in reading initial steam pressure, steam temperature, exhaust pressure, barometric pressure, speed, and power. During the preliminary period readings of all these quantities are to be taken and the calibration and other corrections are to be applied immediately. Whatever adjustment is possible should be made in an effort to set all conditions at the standard values. Very often inattention to the barometric pressure, to its correction for temperature, or to a column of water on the exhaust-pressure U-tube or the steam pressure gage, results in running a test with one or more conditions unnecessarily different from the standard values. Attention must be paid to all such points, and every effort made to run the test with the standard conditions. If tests are made at different loads, the conditions should be as nearly standard as possible for each load, regardless of the possibilities at other loads. For example, standard conditions may be possible at half load and not at full load.

80 In particular, every effort must be made to test the turbine with an exhaust pressure as near as possible to the standard value because corrections for variations in exhaust pressure, especially when extrapolated, are of very uncertain accuracy, owing to extreme sensitiveness of steam consumption to steam congestion in the lower-pressure stages, which is greatly influenced by exhaust pressure.

81 Small auxiliary turbines are often designed and guaranteed for operation with an initial steam pressure considerably below the pressure prevailing in the plant. If there is a valve in the steam line leading to such units, or to a group of them, the initial pressure may be reduced to the guarantee value very readily. If there is no steam valve by which such regulation may be accomplished, the initial pressure may be taken on the turbine side of the throttle valve, so as to give opportunity for pressure control. The pressure to be maintained in such case shall be lower than the standard or guarantee pressure by the amount of the pressure drop through the throttle valve when wide open. It is to be noted that this method of measuring initial pressure applies only to relatively small auxiliary units, and by no means to main units nor to large or important auxiliary machines.

Corrections for Operating Conditions

82 After all efforts have been made, it may still be impossible to have every condition as desired, and the test results must be corrected so as to give, as nearly as possible, the values which would have been obtained with standard conditions.

83 The necessary corrections are given below: Each one gives the percentage change in the steam rate or the total steam for the stated change in the given variable, all other conditions except load remaining constant, as explained in Pars. 100, 101, and 102, that is, all corrections are mutually exclusive.

(*a*) The *Pressure Correction* is the percentage change per 10 lb. per sq. in. The maximum correction shall not be for more than 2 per cent above or 10 per cent below the standard absolute pressure. There is also a superheat correction (Par. 83*b*) for cases of pressure variation when the superheat also varies.

(*b*) The *Superheat Correction* is the percentage change per 10 deg. fahr. change in superheat. The maximum correction shall not be for more than 50 deg. either way. No test with superheat less tnan 25 deg. fahr. shall be run on a turbine designed for superheated steam. For turbines running with high back pressure, the superheat correction becomes very large and is attended with some uncertainty. In such cases the range over which correction may be made must be materially reduced. In the case of tests with initial superheat of 25 deg. fahr. or lower, a throttling calorimeter shall be installed and observed, since it is possible to have coexistent liquid water and moderately superheated steam. Should the calorimeter indicate the presence of liquid water, the accuracy of the test as well as the establishment of suitable corrections, is open to serious question. Efforts should be directed to improving operating conditions rather than to securing corrections. In tests with moderately superheated steam, all drips should be permitted to blow steam because of the fact that water and superheated steam may persist together. With drips blowing freely, there is greater certainty that moisture is not reaching the turbine. When the steam rate of a turbine is determined by weighing or measuring the water fed to a boiler, it will not be possible to permit drips to blow, unless special precautions are taken to determine the quantity of steam and water discharged.

(*c*) *Moisture Correction.* For cases where guarantees are made for wet steam, this correction is the percentage of change in steam rate for each 1 per cent change in initial moisture. The maximum correction shall not

be for more than 2 per cent variation of moisture from standard conditions. Tests with saturated steam (wet or dry) are necessarily of a low degree of accuracy, and should therefore be avoided if possible. Turbines guaranteed for saturated steam are much more accurately tested with slight superheat and if this is done a superheat correction curve is needed.

(*d*) The *Vacuum Correction* is the percentage change per inch of mercury change in absolute exhaust pressure. The maximum correction shall not be for more than 1 inch of mercury if the absolute exhaust pressure during the test is higher than the standard, nor for more than 0.3 inch of mercury if the absolute exhaust pressure during the test is lower than the standard. Different corrections are required for different loads, as half load, full load one and one quarter load, etc.

(*e*) The *Back-Pressure Correction* (for non-condensing turbines) is the percentage change per pound change in back pressure. The maximum correction shall not be for more than 20 per cent of the rated absolute exhaust pressure, nor in any case for a variation of more than 10 lb. per sq. in.

(*f*) The *Speed Correction* is the percentage change for a 10 per cent change in speed. The maximum correction shall not be more than 5 per cent either way.

(*g*) The *Power-Factor Correction* is the percentage change for each 10 per cent change of power factor. The maximum correction shall not be for more than 20 per cent difference of power factor either way.

(*h*) *Load Curve*. A curve must be provided showing total steam, corrected to standard conditions, plotted as ordinates, and actual load in kilowatts, or the like, as abscissas. This is sometimes known as the "Willans line." In case the performance is guaranteed with different combinations of hand valves, the curve must have branches from tests for each combination. The difference between values on this curve, of total steam at rated load (or other standard load), and of total steam at the corresponding actual test load, divided by total steam at rated load (or other standard load), gives the correction to be used as explained for the other corrections.

84 *Methods of Containing Corrections*. Numerical values of the corrections listed above must be obtained for the turbine under test, for each condition differing from the standard value. This shall be done in either of three ways:

a A schedule of corrections may be embodied in the purchase contract

b A schedule of corrections may be furnished by the turbine manufacturer and accepted in writing by all interested parties before the turbine is put into operation.

c Auxiliary tests may be run to determine corrections at the time a regular test is run.

85 *Correction Tests*. The tests of Par. 84(*c*) will consist of runs with each of a series of values of the condition in question, with all other conditions except load as near as possible to standard, and with load set as in Pars. 97 and 98.

86 At least one set of such correction tests shall be run with loads near the rated value, and reduced so as to give correction curves for rated load. Other sets may be required for other loads. This is especially true for the vacuum correction, for which values must always be determined over the complete range of both load and vacuum covered during the steam-rate tests. A correction curve in each case should extend as nearly as possible to the standard value. However, extrapolation by extension of a curve may sometimes be necessary, and in such cases the curve should extend backward a considerable amount in order to fix the correct shape.

87 The correction curve for initial steam pressure may be obtained by throttling, with further correction for changed superheat from a superheat-correction curve, or by proper boiler control. The initial pressure-correction curve on a turbine with valves opening at loads slightly above a standard load should be run with such loads as will put the valves, etc., in the position designed for the standard load as explained in Pars. 97 and 98.

88 The correction curve for superheat may be made by varying the rating at which the boilers are operating. If several boilers are available the superheat may possibly be varied by changing the number of boilers supplying steam to the turbine, and hence the rating of these boilers. Separately fired superheaters may be used. Otherwise superheat is practically uncontrollable. The reduction of superheat by spraying water into the steam line usually is not permissible, because of the difficulty of getting a thorough mixing. Decrease of superheat usually gives but slight decrease of output so that usually a superheat curve may be run at the standard load. However, whenever necessary, the load must be decreased as mentioned in Pars. 97 and 98, so as to put the valves, etc., in the position designed for standard load.

89 The exhaust pressure of a condensing turbine can be raised by admitting air into the condenser or suction line of a dry-vacuum pump by means of a valve whose adjustment gives a means of control. However, the exhaust pressure cannot be lowered below the value attained with the maximum flow of the coldest water available, together with the maximum-capacity operation of the air and condensate removal pumps. In case of a non-condensing turbine, the back pressure may be varied in one direction by throttling with an exhaust valve, and in the other direction by use of a test condenser or by providing a free atmospheric exhaust instead of an exhaust head, or by some similar expedient.

90 An exhaust-pressure, back-pressure, or vacuum curve must be run at the constant standard flow computed from the standard steam rate and load. This curve will of course be run with standard initial steam pressure and superheat so that constant standard flow puts the valves, etc. in the position designed for the standard load, as specified in Pars. 97 and 98.

91 The load curve is usually made as a part of the regular tests and, if so, no special correction curve is needed. There are usually regular tests at a number of loads, all at conditions as near standard as possible, and these are to be used in the manner given later.

92 Tests for the various correction curves usually have slight variation in all of the conditions in addition to the one for which the correction curve is run. A first approximation must be made by assuming corrections for all of the variables, except the one in question, including the load correction. This gives a tentative set of correction curves for rated load as well as the tentative total steam-load curve. From these curves a better set of tentative correction factors is to be obtained and all of the tests worked up again, giving a second set of correction curves for the rated load as well as a total steam-load curve. From this set of curves, another set of curves is to be drawn if necessary, until finally a set is obtained which would not be altered by further trials.

93 The correction tests shall be finally plotted so as to show the test values of steam rate for various values of a variable condition as ordinates, and the variable condition as abscissas. A straight line often represents the points, but sometimes a curve must be drawn. The slope of the straight line or of the chord of a curve (the chord being drawn between test load and standard load) divided by the standard steam rate gives the percentage correction. Often a curve is also drawn with the loads which existed for each correction test (see Pars. 97–98) as ordinates and the variable condition as abscissas.

94 The correction obtained directly from the correction tests reduces conditions to the standard value of the variable, for the load shown by the latter curve at the standard value of the variable. If the valve arrangement specified in Pars. 97 and 98 has been attended to carefully this load will be the standard load, since the curve of load against a variable condition will pass through the point giving standard load at standard value of the variable. However, exactness in this matter is not essential, because percentage correction curves differ but little for wide variations of load. Hence absolute accuracy in setting the standard valve position is not required for correction tests.

95 The general formula for a correction may be developed as follows:

R_s = steam rate at standard conditions
R_t = steam rate at test conditions
V_s = the standard value of one of the variable conditions (e.g., pressure, temperature, etc.)
V_t = the value of the same variable prevailing during the steam-rate test
C = percentage correction for this variable condition

$$C = \left[\frac{R_t - R_s}{V_t - V_s}\right] \times \frac{100}{R_s}$$

If the correction curve is a straight line, the correction C will be constant for all values of R_t and V_t; but if the correction curve is not straight, the ratio of the change in R to the change in V must be taken as the slope of the chord joining the points on the correction curve corresponding to the actual values of V_s and V_t.

96 The total correction is a combination of the separate corrections for the several variable conditions. Let

C_p = percentage correction for the actual deviation of initial pressure from the standard value

C_t = the same for initial temperature
C_x = the same for exhaust pressure

and so on for all variable conditions. Since each of these is expressed as a percentage of the steam rate at standard conditions, the following relation holds:

$$R_s\left(1+\frac{C_p}{100}\right)\left(1+\frac{C_t}{100}\right)\left(1+\frac{C_x}{100}\right)\ldots = R_t$$

This may be solved for the value which the steam rate would have were all conditions standard

$$R_s = \frac{R_t}{\left(1+\frac{C_p}{100}\right)\left(1+\frac{C_t}{100}\right)\left(1+\frac{C_x}{100}\right)\ldots}$$

If the various corrections are small, a close approximation is given by the following equation:

$$R_s = \frac{R_t}{1+\frac{C_p+C_t+C_x+\ldots}{100}}$$

This, however, is not strictly accurate; wherefore, in all cases, the several corrections shall be combined by multiplication, and not by addition.

97 *Selection of Load to Correspond to a Condition below Standard.* In running the regular test, when any one of the conditions is unavoidably below the standard value, or in running auxiliary tests for corrections, with any condition purposely set below the standard value, it is important to select a load correspondingly low in order to avoid operation of the turbine in a region which will introduce irregularities other than that corresponding to the single changed condition.

98 For instance, if a turbine has hand or automatic valves of any kind designed to open at a load slightly above the rated load, then operation at the rated load when there is any condition below standard would make the turbine function in a manner which was not contemplated for rated load. Hence, a load always must be set so as to put the valves, etc., in the position designed for the standard load with all conditions standard. This will give a lighter load, with any condition below standard. The correction for variation of any one condition is usually the same for a fairly wide variation of load, so that so far as a correction itself is concerned, it is not important to hold an exact load. For all of these reasons, correction tests should usually be run at constant valve position, and not at constant load. The standard position of the valves need not be set with absolute accuracy, because the correction for difference of load is readily given by the load curves.

99 Such operation with valves, etc. functioning in the manner designed for standard load may be accomplished as described in Pars. 67, 69 and 70.

100 *Use of Corrections.* Each correction may be applied either to the total steam or to the steam rate.

101 For each correction there are given the maximum limits for which it should be used. Usually all corrections, except the vacuum and speed corrections, apply to any load between 50 and 125 per cent of the rated load, and this is to be assumed unless there are tests or specifications to the contrary. The vacuum and speed corrections do not apply over any material range of load.

102 In each case percentage corrections are computed on the basis of the performance at the conditions specified as standard, which is taken as 100 per cent. This means that the net percentage correction for any item of variation is to be subtracted from or added to unity and the result used as a divisor for the test value of total steam or steam rate to obtain the corrected value. Such a corrected value gives the total steam or steam rate corrected to the standard value of the condition in question, for the standard load as explained in Par. 93. There must be made a further correction for load as specified by the last item, if the test load differs from the standard load. The several divisors giving the corrections for each condition which varies are to be multiplied together and used as a single divisor for the test value of total steam or steam rate to obtain the corrected value.

EXAMPLE. The following hypothetical example will serve to illustrate the method.

Initial steam pressure is low, making the actual steam rate 2.1 per cent higher than it would have been with standard initial pressure. Correction divisor = 1.021.

Initial superheat is high, making the actual steam rate 2.9 per cent lower than it would have been with standard superheat. Correction divisor = 0.971.

Exhaust pressure is high, making the actual steam rate 1.8 per cent higher than it would have been with standard exhaust pressure. Correction divisor = 1.018.

The load is low, making the actual steam rate 0.4 per cent higher than it would have been with standard load. Correction divisor = 1.004.

The combined correction divisor will be

$$1.021 \times 0.971 \times 1.018 \times 1.004 = 1.013$$

That is, with all things considered, the actual steam rate is 1.3 per cent higher than it would have been with all conditions conforming to the established standard.

Actual steam rate = 1.013 × standard steam rate

$$\text{Standard steam rate} = \frac{\text{Actual steam rate}}{1.013}$$

CALCULATION OF RESULTS

103 *Correction of Total Steam or Steam Rate.* The observed total steam or steam rate, for each constant load test run, shall be corrected as follows:

a For condenser leakage (see Instruments and Apparatus, Pars. . . . . , for methods of determining leakage). If this leakage is measured by a continuous method (chemical or electrical), the observed total steam or steam rate for each constant-load test run shall be reduced by a percentage which is the average result of not less than three leakage determinations conducted during that same constant-load test run. If leakage is determined by either of the direct weighing methods, the observed total steam or steam rate for each constant-load test run shall be reduced by a percentage which is the average result of the two leakage tests between which the constant-load run is conducted.

b For water other than condensate and condenser leakage entering the condenser, and for condensate which does not pass to the measuring or weighing tanks, see Pars. 28 to 34 of this code.

c For operating conditions, as provided in Pars. 73 to 90 of this code.

104 If the turbine is tested at several loads covering an appreciable range, the total steam, in pounds per hour, shall be plotted against the average turbine load for the respective runs, and the total steam curve shall be drawn. The curve of steam rate, in pounds per unit output, against load, shall be computed point by point from the *curve* of total steam against load, and not from the actual observed values. For purposes of comparison, the values of steam rate computed from the observations shall be plotted also.

105 *Heat Rate.* For turbines operating on the regenerative or reheating cycles, the steam rate, in pounds per unit output becomes of small value for comparative purposes. In such cases it is necessary to compute the heat consumption, in B.t.u. per unit output, by multiplying the steam rate, in pounds per unit output, by the following:

Heat content of steam at the throttle
minus
Heat content of steam at point of departure from the turbine for reheating (one or more values)
plus
Heat content of steam at point of return to the turbine after reheating (one or more values)
minus
Heat content of water at the temperature at which it leaves the turbine to return to the boiler room. In the case of the simple single-flow turbine, this shall be taken as the temperature of the boiling point corresponding to the absolute pressure prevailing at the turbine exhaust flange.

106 *Steam Cycle Efficiency.* The efficiency of an ideal turbine operating under the same conditions as the actual turbine shall be computed as outlined in the "Code on Definitions and Values."

107 *Engine Efficiency.* As defined in Definitions and Values, Par. . . . . , the engine efficiency of the actual turbine is the ratio of the steam rate or the heat consumption of the ideal turbine to that of the actual turbine. For turbines operating on the regenerative or re-

heating cycles, only the heat consumption ratio has meaning, since the steam rate alone does not express the economy of the unit.

108 *Regulation.* See "Test Code for Speed-Responsive Governors."

DATA AND RESULTS

109 The data and results should preferably be reported in accordance with the form in Table 1 appended hereto, with additions or omissions in conformity with the object of the test. A brief report may be made up of the data included in Table 2. Each item in these tables will frequently consist of a series of values, for various conditions, such, for example, as steam rate at various loads.

110 If steam-rate tests have been run at several loads the principal results shall be shown in curve form. Such curves shall all be plotted with a common scale of load as abscissas, with the following ordinates:

a Total steam, to as large a scale as possible
b Steam rate, or heat consumption
c Thermal efficiency
d Engine efficiency
e Such stage steam pressures as have been observed (see Par. 58). For a turbine governed by throttling, as long as the overload by-pass valve remains closed, these curves should be straight lines, similar to the Willans line, provided the initial steam conditions and exhaust pressure do not change greatly. Any marked departure from this straight line is a strong indication of erratic results, and should call for a verification of the points in question by further testing. An added use of this curve is in checking the internal condition of the turbine. The position and slope of this line vary with the mechanical condition of the turbine, and often reveal circumstances calling for prompt investigation.

111 As a check on operating conditions, the report should present for each test run graphic logs, with time as abscissas, and with the following ordinates:

a Initial steam pressure
b Initial steam temperature
c Exhaust pressure
d Load
e Rate of flow of condensate.

112 See "Code on General Instructions," for detailed instructions for plotting curves.

TABLE 1 DATA AND RESULTS OF STEAM-TURBINE {STEAM-RATE / CAPACITY} TEST

(*A.S.M.E. Test Codes of 1923*)

GENERAL INFORMATION

(1) Date of test............
(2) Location............
(3) Owner............
(4) Builder............
(5) Test conducted by............
(6) Object of test............

DESCRIPTION, DIMENSIONS, ETC.

(7) Type of turbine (impulse, reaction, or combination)
a Number of stages............
i Impulse
Note which impulse wheels, if any, have two or more "velocity" stages
ii Reaction (pairs of rows, one stationary, one moving)
b Condensing or non-condensing............
c Type of governor............
d Rated speed............
(8) Turbine rating:
a Maximum continuous rating of generator............
b Load at best economy............
c Maximum overload capacity of turbine............
d Power factor............
(9) Class of service (driving of electric generator, centrifugal pump, etc.)............
(10) Driven unit:
Type, size, rating, etc., of driven unit............
(11) Auxiliaries:
a Condensing equipment............
i Type............
ii Make............
iii Rated capacity............
b Oil pumps............
i Type............
ii Drive (direct or independent)............
c Auxiliaries to driven unit............
Include under this head a statement of size, type, drive, etc., of exciters, ventilating fans, oil pumps, etc., auxiliary to the driven unit.

RESULTS

(12) Run number............
(13) Duration............hr.
(14) Average load............hp. or kw.
(15) Initial steam pressure............lb. per sq. in. abs.
(16) Initial steam temperature............deg. fahr.
(17) Exhaust pressure..lb. per sq. in. or in. Hg. at 32 deg. fahr., abs.
(18) Temperature of condensate returned............deg. fahr.
(19) Actual steam rate............lb. per hp-hr. or lb. per kw-hr.
(20) Steam rate corrected to guarantee conditions............lb. per hp-hr. or lb. per kw-hr.
(21) Guarantee steam rate............lb. per hp-hr. or lb. per kw-hr.
(22) Heat consumption, acutal..B.t.u. per hp-hr. or B.t.u. per kw-hr.
(23) Thermal efficiency, actual............per cent
(24) Engine efficiency, actual............per cent

SUPPORTING DATA

Output:

(25) Output, hp-hr., kw-hr., ft-lb., or the like, if observed independently............
(26) Average load during run, hp. or kw. This may be observed directly, Item (25) being omitted............
(27)
(28) Additional items covering a turbo-generator will be added after sub-committee No. 19 (Instruments and Apparatus) has prepared detailed instructions for the measurement of the output of such units with their various more common arrangements of auxiliaries, such as exciters, fans, and oil pumps. Meanwhile, several item numbers are left blank.
(29)

Atmospheric conditions:

(30) Barometric pressure:
a In. of mercury at 32 deg. fahr............
b Lb. per sq. in. abs............
(31) Room temperature............deg. fahr.

Condition of Steam Supplied, Observed Just Ahead of the Throttle:

(32) Pressure:
a By gage............lb. per sq. in.
b Absolute............lb. per sq. in.
(33) Temperature............deg. fahr.
(34) Superheat............deg. fahr.
(35) Moisture............per cent

Condition of Steam in Turbine Stages:

(36) Stage *xy*. Pressure:
a By gage....lb. per sq. in., or in. of mercury at 32 deg. fahr.
b Absolute......lb. per sq. in., or in. of mercury at 32 deg. fahr.
(37) Stage *xy*. Temperature......deg. fahr. or Moisture....per cent

Condition of Steam Rejected:

(38) Pressure:
a By gage (vacuum or pressure)....lb. per sq. in. or in. mercury at 32 deg. fahr.
b Absolute......lb. per sq. in. or in. mercury at 32 deg. fahr.

Condensate Returned to Boiler Feed System:

(39) Temperature............deg. fahr.

Speed:

(40) Speed............r.p.m.
(41) Variation in speed between no load and full load:
a ....r.p.m.
b ....per cent of rated speed

Thermal Data:

(42) Steam tables used............
(43) Steam supplied:
a Heat content............B.t.u. per lb.
b Entropy............units per lb.
(44) Steam leaving to be reheated: heat content........B.t.u. per lb.
(45) Steam returned after reheating: heat content.....B.t.u. per lb.
Repeat items (44) and (45) for each stage of reheating.
(46) Temperature of boiling point corresponding to exhaust pressure (Item (38))............deg. fahr.
(47) Heat content of water at temperature of item (46)..B.t.u. per lb.
(48) Heat content of water actually returned to boiler-feed system (temperature of Item (39))............B.t.u. per lb.
(49) Heat supplied per pound of steam............B.t.u. per lb.
(50) Net work per pound of steam developable in ideal turbine operating under the same conditions as the actual turbine............B.t.u. per lb.
(51) Thermal efficiency of the ideal turbine............per cent

Corrections for Operating Conditions:

(52) Guarantee conditions:
Initial pressure..........................lb. per sq. in. abs.
Initial superheat or quality................deg. fahr. or per cent
Exhaust pressure....lb. per sq. in. abs., or in mercury at 32 deg. fahr. abs.
Speed..r.p.m.
Power factor......................................per cent
Load...hp. or kw.

(53) Corrections from guarantee to test conditions, as percentages of guarantee values:
a For initial pressure.....................................
b For initial superheat or quality..........................
c For exhaust pressure....................................
d For speed..
e For power factor..
f For load...
g Total correction for operating conditions.................

Correction for Water Gained and Lost:

(54) Condenser leakage (state how determined, and give pertinent data)..............lb. per hr., or per cent
(55) Other water gained or lost (give complete information, including method of determining quantities)..........lb. per hr.

Steam Quantities:

(State method of determining total steam.)
(56) Total steam observed during run...........................lb.
(57) Total steam per hour.................................lb. per hr.
(58) Total steam per hour, corrected for water gained and lost...lb. per hr.
(59) Total steam per hour, further corrected for operating conditions..lb. per hp-hr. or lb. per kw-hr.
(60) Steam rate, actual test........lb. per hp-hr. or lb. per kw-hr.
(61) Steam rate, actual test, corrected for water gained or lost............lb. per hp-hr. or lb. per kw-hr.
(62) Steam rate corrected to guarantee conditions......................lb. per hp-hr. or lb. per kw-hr.

TABLE 2 NOTES AND CALCULATIONS

(Numbers in parentheses refer to Items in Table 1 (pages 767–768), except where specifically designated as referring to paragraphs of the code text.)

(1) to (24) Comprise a brief summary, and may be reported without the subsequent supporting material.
(7) to (11) Call for the description of main and auxiliary equipment. They should be expanded to include all name-plate data, and such other facts as may be required to describe the units completely. The facts specifically mentioned are in the nature of an indispensable minimum, which it is highly desirable to amplify. In particular, the manufacturer's serial number and the user's serial number or other identification should be given for each piece of apparatus.
(12) For this and all subsequent items, there will appear a column of values for each test run.
(13) See Par. 74
(14) Same as (26)
(15) Same as (32) *b*
(16) Same as (33), or, if appropriate, substitute (35)
(17) Same as (38) *b*
(18) Same as (39) for a turbine with regenerative water heating by extracted steam. For a turbine without regenerative heating, (18) is taken the same as (46).
(19) Same as (61)
(20) Same as (62)
(21) Taken from contract
(22) $= (61) \times (49)$
(23) $= 2545/(22)$ or $3413/(22)$
(24) $= 2545/(61) \times (50)$ or $3413/(61) \times (50)$
$= (23) \times (49)/(50)$
(25) Output may be measured directly by integrating instruments.
(26) Load may be measured directly by dynamometer. See Par. 18, ff.
(30) See Par. 46.
(32) See Par. 36, ff.
(36) and (37) These items will appear for each stage for which observations are made. For a simple single-flow turbine, all such data are optional, but desirable. For turbines operating on the regenerative or reheating cycles, it is necessary that data be given for each stage from which steam is withdrawn, either for reheating or for water heating, and for each stage into which steam enters after reheating. These items are to be repeated for each stage recorded with the designation of respective stages appended to the item numbers. See Pars. 40 and 58.
(38) See Par. 41, ff.
(39) Is not required for a simple single-flow turbine. For a turbine operating on the regenerative cycle, it is essential that this temperature be observed accurately. See Par. 50.
(42) Tables or charts may be used, but the source of all thermal data must be explicitly stated.
(43) Is taken for the pressure (32) and the temperature (33), or the superheat (34), or the quality (35).
(44) and (45) Are taken for the conditions given in (36) and (37), and like them, are to be repeated for each stage of reheating.
(46) and (47) Are for a turbine operating without regenerative water heating with extracted steam.
(48) Is for a turbine operating on the regenerative cycle.
(49) $= (43)\ a - (44) + (45) - (48)$ For a turbine operating without regenerative water heating by extracted steam, substitute (46) for (48). See Par. 105.
(50) and (51) See "Code on Definitions and Values," Par. ....., for instructions for calculating this quantity for the conditions under which the turbine is operated.
(52) See Par. 84.
(53) See Par. 82, ff.
(54) See Par. 26
(55) See Par. 28, ff.
(56) The method of determining total steam shall be stated. See Par. 23, ff.
(57) $= (56)/(13)$
(58) $= (57)$ corrected for (54) and (55)
(59) $= (58)/(53)\ g$
(60) $= (57)/(26)$
(61) $= (58)/(26)$
(62) $= (59)/(26)$.

Transportation in 1930

According to the author, the outstanding feature of the present-day requirement for modern transportation by the public is speed. He quotes the following paragraph as covering the situation:

> The average American wants to be jerked up from his forty winks, thrown by automatic, mechanical action, from his In-A-Door bed into a cold plunge, slip into his clothes, gulp a "cup of coffee" as he snaps on his tied bow tie, peck at his wife's bob, slide into a container, touch a button, be shot, listening to the morning news by radio, direct into his office chair, coat and hat off, the time between his "G'bye dear" in the perch which he calls "home" and "Call London, Miss Smith," to his secretary being as much less than "nothing" as it possibly can be.

Seventy-five miles per hour seems to be the minimum speed tolerable to Americans of 1925, and in five years hence this figure will be about 90 to 100 miles. Five years hence the same type of vehicles used now will still be used for mass transportation and the greatest group of the public will be carried by electric-railway cars which will be supplemented by eight-wheel, rubber-tired, electric-motor-driven trucks self-propelled, and mechanically driven buses.

The trend of the development of the latest closed automobile design, particularly in Europe, is the perfect streamline vehicle, and the president of a well-known American automobile concern predicts that within a few years the automobile will develop into a small, inexpensive vehicle suitable for one or possibly two passengers, so that each member of a family will operate his or her own unit. One of the younger British scientists has a theory that the future will bring forth small, high-powered automobiles capable of a speed of 100 m.p.h., equipped with wings or some device enabling the operator to fly at will. One can readily visualize driving from a city or congested district until the country highways are reached, then sailing across country at 100 m.p.h. to the destination, or if the latter happens to be in another city, dropping gradually to the open road and then driving the car as one does now within four or five blocks from where one wishes to go.

As the traveling public is always interested in something novel, considerable interest has been aroused by the experiment soon to be tried by the Grand Rapids Railway of Grand Rapids, Mich., with three new electric-railway vehicles built by three different car builders, which are to be light weight and noiseless, being equipped with various new appliances and containing new features to get as far away as possible from "dyed-in-the-wool" street-railway practice, and approach closely to a bus or closed automobile, except of course that they are to be run on tracks and to be operated electrically the same as any other street car.

From facts the author passes to fancy and describes not only Japan-America air transportation with a schedule of 90 hours, but even an air robbery of a postal plane bearing the registered mail. (Paper was written in collaboration by M. J. Oswald, F. L. Markham, and B. W. Frauenthal, the last named being general traffic agent of the United Railways Co. *Official Proceedings of the St. Louis Railway Club*, vol. 30, no. 2, June 12, 1925, pp. 25–36, *g*)

Engineering and Industrial Standardization

Coöperative Relations between A.E.S.C. and Sponsors

THE sponsors for the sectional committee which is to unify the present standards for cast-iron pipe have inquired of the American Engineering Standards Committee whether it would be feasible for them to station the secretary of the Sectional Committee on Cast-Iron Pipe in the office of the A.E.S.C.

It is proposed that all the expenses, including the salary of the secretary, mimeographing, postage, etc. should be paid by the groups maintaining him, and it is proposed that he would be solely responsible to the sponsors and the chairman of the sectional committee, having only liaison relations with the A.E.S.C.

A special committee of the A.E.S.C., under the chairmanship of A. W. Whitney, made a careful study of this whole question in view of the discussion of the subject at the May meeting of the A.E.S.C. Executive Committee. In accordance with the recommendation of the Special Committee the A.E.S.C. Main Committee at its meeting of June 11 unanimously passed the resolution:

Resolved: That a sectional committee, if it so desires, may station a representative in the office of the American Engineering Standards Committee, in order to avail itself of the facilities of the A.E.S.C., provided that all expenses connected therewith shall be assumed by the sponsor of the project in question and provided that the relationship shall be under the surveillance of the Main Committee, and may be terminated by the Main Committee at any time if in the opinion of the Main Committee such relationship shall prove undesirable, and provided, furthermore, that each such application must be acted upon and approved by the Main or Executive Committee. It is to be understood, on the other hand, that the representative of the sectional committee shall be chosen by the sponsor, shall be responsible only to the sponsor, and that the surveillance of the Main Committee shall be limited to obtaining the information necessary to a decision as to whether such an arrangement shall cease or continue.

At the suggestion of the A.S.M.E. Standardization Committee and with the permission of the A.E.S.C. office, we reproduce below a communication which Dr. P. G. Agnew, secretary of the A.E.S.C., addressed to the Main Committee on June 9:

To THE MEMBERS OF THE A.E.S.C.:

This memorandum is prepared in accordance with instructions of the Executive Committee (Minute 1396).

At the May Meeting of the Executive Committee a communication was received (MC 395) from Mr. C. L. Warwick on behalf of the Steering Committee of the sponsors of the sectional committee on Cast-Iron Pipe (viz. New England Water Works Association, American Gas Association, American Society for Testing Materials, American Water Works Association), inquiring whether it would be feasible to station the secretary of the sectional committee in the office of the A.E.S.C., provided the expenses (including the salary of the secretary, mimeographing, postage, etc.) were paid by the groups maintaining him, and provided he were solely responsible to the sponsors and the chairman of the sectional committee, having only liaison relation with the A.E.S.C.

This proposal is closely related to the important question of the stationing of liaison representatives in the A.E.S.C. office by associations, which has been discussed from time to time—see the 1924 Year Book, pages 18 and 19, from which the following is quoted:

It may well be that these tendencies in the use of technical staffs by associations will develop further, and that some of the stronger associations will station a staff representative in the A.E.S.C. office as a permanent liaison representative, as is now being done by trade associations in Germany. In connection with its relation to the A.E.S.C., one association now has under consideration the assignment of a staff engineer to duties similar to those outlined. The Bureau of Standards and the Federal Specifications Board jointly have maintained a liaison representative in the A.E.S.C. office for two years.

I believe it extremely desirable that eventually arrangement be made for the stationing of a dozen or more such liaison representatives of associations, in the A.E.S.C. office. Such an arrangement has been developed in Germany where it has worked out exceedingly well. I believe that at the present time there are thirteen such representatives in the central office. The simple, direct interchange of information and coöperative relation which would follow from such an arrangement would be of the greatest possible value in the work. Such men reporting to and responsible to the organizations appointing them, would naturally enjoy a degree of confidence which would not be possible were they employees of the central organization. Such a plan could also be more readily financed.

A considerable part of the routine work of the A.E.S.C. office could well be done by such men, particularly such matters as the detailed work in connection with the organization of conferences, questions in connection with the submission and approval of the personnel of sectional committees, the systematic interchange of information on progress of work between sponsors, working committees, and other interested bodies, and similar matters of importance to their respective groups.

The services of Dr. Harper and Dr. McAllister, as liaison representatives of the Bureau of Standards and the Federal Specifications Board, have made it possible to carry on the A.E.S.C. work on a scale and with an effectiveness which would have been out of the question had their services not been made available by the Bureau of Standards, though conditions have made it possible for these gentlemen to carry a larger share of the A.E.S.C. work than would normally be the case with association representatives.

Though similar in some respects, the problem is somewhat different in the case of the secretary of a sectional committee, and greater care should be taken in safeguarding the relations between the A.E.S.C. and sponsors, which have been so carefully laid down in the Constitution, Rules of Procedure, and By-Laws. It is a question which we in the office had not thought of until the proposal was made by the Steering Committee of the sponsors for the project on Cast-Iron Pipe.

In this case it is understood that considerable difficulties exist, rising out of the relations within the producing group, and past relations between the producing group and the major consuming group. It has been planned by the sponsors that an eminent consulting engineer, of New York, be chosen as chairman, and they wish to put at his disposal the services of a competent engineer to act as secretary of the sectional committee, under the immediate direction of the chairman of the sectional committee, and the general authority of the sponsors. It is believed by the Steering Committee of the sponsors that the work can be carried out very much more economically and effectively if this engineer has his desk in the A.E.S.C. office. On the one hand, there would be a considerable saving of expense compared with opening a separate office, and on the other hand, direct contact with the information files and the general files of the office would be a great advantage both from the technical and the administrative point of view, particularly since this project is an extensive one and touches a large number of other lines of work.

Financial Arrangement. If such an arrangement is entered into, the financial and other administrative arrangements should be very definite and clear-cut. It seems to me that the salary of the secretary should be paid directly by the sponsors, as should also all bills for mimeographing, stationery, etc. and the salary of the stenographic assistance should be paid directly, or prorated on an equitable basis. I should be inclined to think that the desk room might well be furnished by the A.E.S.C., with the understanding that the A.E.S.C. should be relieved of all details in connection with the project which would normally fall upon the A.E.S.C. office. Also the services of the secretary of the sectional committee should be available to the A.E.S.C. for "pinch hitting" work at times of conferences and meetings, and in emergencies.

As the members of the A.E.S.C. are well aware, the pressure on our budget is uncomfortably great, and until more funds are available it is not feasible to ask for an adequate increase in the budget. Under the circumstances, I suggested to the Steering Committee that if the plan should be approved by the A.E.S.C., it might be feasible to arrange for Mr. Gaillard to give half of his time to the Sectional Committee and the other half of his time to the A.E.S.C. I believe such an arrangement would be of great mutual advantage.

Safeguards. As was pointed out at the meeting of the Executive Committee, any such arrangement from the point of view of either association liaison representatives (in which I am decidedly more interested) or sectional committee secretaryships, should be surrounded by adequate safeguards. For example:

1 The secretary of the sectional committee should be engaged by the sponsors, and be responsible only to them and to the chairman of the sectional committee.

2 This responsibility should be definitely indicated in his title, on stationery, and in correspondence generally.

3 The financial arrangements should be such that no association or sectional committee expense should fall upon the A.E.S.C. and normally should be such as to result in a definite economic advantage to the A.E.S.C.

4 Applications for such arrangements should be approved in each case by the Main or the Executive Committee.

5 Either the Chairman of the Main Committee, or a special committee, should make a personal examination into the workings of any arrangement that may be entered into, and should report their findings to the Main Committee from time to time.

With such safeguards thrown around, I believe it would be extremely desirable that all possible efforts be made to bring about the stationing of liaison representatives in the A.E.S.C. office, by coöperating associations. I believe that such an arrangement would bring about a much wider recognition of the A.E.S.C. work, and of both moral and financial support of it on the part of industrial groups and I hope that as an experiment in this direction, the request of the sponsors for the cast-iron-pipe project will meet with the approval of the Main Committee.

P. G. Agnew,
Secretary.

Correspondence

Contributions to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities or policies of the Society in Research and Standardization.

Efficiency of More than 100 Per Cent

To the Editor:

Referring to the mention of the Brünler boiler on page 381 of the May issue of Mechanical Engineering, I should like to draw attention to articles which appeared in the Journal of the Royal Dutch Institute of Engineers as long ago as 1914–1915 and in which it was made clear that (see *De Ingenieur,* 1914, pp. 935–942; and 1915, pp. 11–12 and 29–30):

1 In 1894 the Dutch engineer Huet patented a subaqueous-combustion boiler, which was, however, not developed commercially.

2 In 1895 Mr. Brünler developed his idea of subaqueous combustion.

3 Not until about 1908 was an experimental installation built.

4 In 1914 an industrial installation was in use for evaporating water from chemical solutions.

5 The outlook for a more general application of the principle involved was not at that time favorable.

6 It is not necessary to have recourse to ultra-violet rays or to the breaking-up of atoms to explain the extra evaporation from a solution of sodium salts, etc. as the products of combustion, combining chemically with part of the dissolved matter, furnish a considerable amount of heat which cannot be considered as being part of the calorific value of the fuel.

D. Dresden.[1]

The Hague, Holland.

[1] Professor of Mechanical Technology, Institute of Technology, Delft.

The Scientific Placement of Young Engineers

To the Editor:

From time to time we hear criticisms of our own and the other leading engineering societies to the effect that they do not interest themselves sufficiently in the financial and business advancement of their members.

Personally I am of the opinion that the practice of our Society in confining its activities mainly to the development of the science and art of engineering, and in leaving it to other organizations to deal with the commercial welfare of engineers, is in the main correct. At the same time it has long been recognized that in all engineering activities there are two factors, the man and the machine, and as in the end the machine is only a tool for enlarging the activities of the man, we have now accepted the science and art of industrial management as a legitimate part of the field of engineering.

Although much thought has been given by our Society to questions relating to the scientific management of men, I am of the opinion that not enough thought has been given to the scientific placement of men, and especially of young engineering graduates. This view is based upon personal experience.

For the last eighteen months I have been engaged in volunteer work for the Society in interviewing candidates for membership residing in my neighborhood, and during this time I have interviewed over one hundred men, a majority of whom are under thirty years of age.

Now these interviews have not merely consisted of ten-minute talks, for, as many of the men are not at liberty during business hours I have generally asked them to visit me at my home, where they often stay two or three hours telling me of their business troubles and ambitions; and incidentally I have learned not only that the young men have all the troubles in life, but that there are other things which are of far greater importance to those interested in the welfare of our profession.

To begin with, very few young men are satisfied with the line of work in which they are now engaged and are willing to consider it as their life work.

Although we all recognize the fact that the other fellow's job always appears more rosy than our own, when one attempts to advise these young men he is confronted with the fact that it is not only very difficult to form an adequate opinion as to the possibilities of his own line of work, but that reliable data and unbiased opinions as to the opportunities in other lines are practically unavailable—all of which is causing a great economic loss and hardship.

The field of engineering is so wide—and is growing wider every day—that the young man leaving college is simply bewildered with the multiplicity of choices open to him, and there appears to be no place to which he can turn for reliable information. Although it is always difficult for the beginner to decide wisely, the engineering graduate, owing to his special training, should be in a better position to judge of opportunities correctly than most young men of his age if reliable data were available for him, and it appears to me that the work of collecting such data and its proper presentation is a matter which should be of interest to our Society, for it is these young men who will form our Society in the future.

I do not suggest that this work should be carried on at the expense of the Society but that our attitude toward it should be similar to that we have taken with regard to the advancement of scientific management, of which it might be considered to be a legitimate offshoot.

I am of course aware that a certain amount of such placement work has been done, but I consider it a major crime to force a man in a direction in which he does not want to go just because some one thinks that he ought to go that way; and I believe that it is

a duty that the engineering profession owes to its young men to make available to them as complete data and information as possible, upon which they may base a decision as to their life work; and as before suggested, I am convinced by my experience that this is not merely the expression of a pious hope, but a clear statement of a very pressing need.

This work would require the expenditure of a certain amount of money, but it offers great possibilities of savings to industry at large. It is not work which can be left in the hands of immature efficiency experts, but in the hands of men with a wide experience of life and broad sympathies, the possibilities appear to be almost endless.

In connection with this, I am reminded of a statement by an acquaintance who is at the head of an engineering school, that in his opinion the baking of bread is a heat-treating problem pure and simple, and is just as much an engineering problem as the heat treatment of steel—and after all, why not?

If this view is correct, it opens up a vast field of research as to the possibilities of employment for technical graduates in businesses, which are not at present considered to be of an engineering nature.

It is, I think, a question which colleges, engineering societies, and all those who have the welfare of our future engineers at heart should be interested in, and surely there must be other members of the Society to whom it will appeal.

James O. G. Gibbons.[1]

Newark, N. J.

[1] Consulting Engineer. Mem. A.S.M.E.

A.S.M.E. Boiler Code Committee Work

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of the Society, for approval, after which it is issued to the inquirer and simultaneously published in Mechanical Engineering.

Below are given interpretations of the Committee in Cases Nos. 481, 484, 486, 495, 496, and 497, as formulated at the meeting of June 5, 1925, all having been approved by the Council. In accordance with established practice, names of inquirers have been omitted.

Case No. 481

Inquiry: In calculating the strength of the rivets in shear in reinforcements for openings larger than 3-in. pipe size, under Par. P-261 of the Code, is it necessary to figure one-half of all the rivets in the reinforcement, or should the first two rivets on the horizontal center line be deducted and then one-half of the remainder figured?

Reply: It is the opinion of the Committee that the application of Par. P-261 to such a reinforcement as described will necessitate that the first two rivets on the horizontal center line must be deducted and that one-half of the remaining rivets shall be used for the calculation of the strength of rivets in shear.

Case No. 484 (Reopened)

Inquiry: Is it permissible to connect a low-water signal or alarm of the whistle type to the water-column or water-gage connections of a boiler?

Reply: It is the opinion of the Committee that Par. P-295 of the Code does not permit the connection or attachment of any form of apparatus to the pipes connecting a water column to a boiler that may under any conditions allow steam to escape therefrom. This applies only to devices or attachments other than those enumerated in Par. P-295, and it is not to be construed that it will prohibit the use of feedwater regulators, damper regulators, or high- and low-water alarm devices in which there is no appreciable flow of steam from the connecting line.

Case No. 495

Inquiry: Is it permissible under the rules in the Code to use universal or edge-rolled boiler plate for butt straps in view of the tension-test requirements specified in Par. P-190? It was pointed out that in a previous interpretation of the Boiler Code Committee the application of this tension-test requirement in Par. P-190 had been made to apply to either dimension of the plate as might be desired, but that there is still some uncertainty as to the acceptability of the so-called universal or edge-rolled plate under the present revised edition of the Code.

Reply: There appears to be nothing in the Material Specifications Section of the Code that pertains specifically to universal or edge-rolled boiler plate. It has been proposed to revise the Code to provide for the use of this material for double butt straps by the addition of the following:

Make present Par. S-11 section *a* and change side head to read:

Bend Test for Sheared Plates

Add new section *b* to read:

Bend Test for Universal or Edge-Rolled Plates where Permitted for Double Butt Straps. b. The bend-test specimen shall withstand being bent cold through 180 deg. without cracking on the outside of the bent portion as follows: For material 1 in. or under in thickness, around a pin the diameter of which is equal to $1^1/_2$ times the thickness of the specimen; and for material over 1 in. in thickness, around a pin the diameter of which is equal to 3 times the thickness of the specimen.

Case No. 496

Inquiry: Is it the intent of Par. U-68 that the allowable stress of 5600 lb. per sq. in. shall refer to the normal thickness of the shell plate, or to the thickness at the welded joint, which under Par. U-71 is required to be from 20 to 50 per cent thicker than the plate?

Reply: It is the opinion of the Committee that the allowable stress specified in Par. U-68 shall be calculated on the basis of the minimum thickness of the shell plate.

Case No. 497

Inquiry: Is it not the intent of the marking requirements in Par. U-66 of the Code that it shall be optional for the manufacturer to stamp directly on the shell plate, or solder a non-ferrous name plate to the shell plate, as is provided for in Par. P-332 of the Power Boiler Section? With tanks having thin shells it is very difficult to apply the marking by stamping.

Reply: It has been the intent of the Committee to provide for the marking of unfired pressure vessels along such lines as will conform in general to the corresponding requirement in the Power Boiler Section of the Code, and it is the opinion of the Committee that to mark pressure vessels by use of a non-ferrous name plate, brazed, or otherwise irremovably attached to the shell plate, will conform to the spirit of the requirement in Par. U-66.

Errata

In Professor Eckart's paper on the specific-heat-specific-gravity-temperature relations of petroleum oils which was published in the July issue of Mechanical Engineering, the coefficient in the last formula on page 538 should read 0.000823 instead of 0.00823. Further, in the last quotation on the same page, the bracketed comment is that of the writers quoted, and not that of Professor Eckart.

In the August issue of Mechanical Engineering there was published a paper by S. Einstein on Standardization of Machine Tools, pages 614–618. This paper should have been credited to the Cincinnati Local Section of the A.S.M.E. before which it was read on May 15, 1925.

Standardization in the Machine-Shop Industries

By CARL J. OXFORD,[1] DETROIT, MICH.

MUCH has been said and written of the desirability of standardization, and it is generally conceded that it should be extended to an ever widening field. However, when we begin to put standardization into actual practice in a large field, such as the machine-shop industries, we almost invariably meet with the obstacles of current individual practices or designs, and the very human tendency to oppose changes unless they are of immediate and obvious advantage.

The purpose of this article is to emphasize some of the greater advantages of standardization as applied to the machine-shop industries, thus adding weight to the work undertaken by the American Engineering Standards Committee, and to summarize briefly the various standardization projects completed, underway, or contemplated by the latter body and others.

The expression of individuality, if opposed to standardization, may perhaps have some aesthetic values, but is invariably both useless and expensive. Real individuality in the field here under consideration is expressed rather in greater utility, in correlation to other necessary units or conditions, in simplification, and in the ultimate reduction of costs.

Competitive conditions, local, national, and international, will eventually dictate the simplification and unification of elements and industrial equipment to a point where costs are reduced to the lowest quantity, and all unnecessary waste in time, effort, and material is eliminated.

Frequently it is difficult to get a long-range view of the ultimate advantages of standardization and at times it may seem inadvisable because it involves immediate expenditures without correspondingly immediate returns. It should be borne in mind, however, that the longer standardization is delayed the more costly it usually becomes and the longer one must be without its advantages.

Consider for a moment the matter of drawings and specifications both of equipments used and of articles or machines manufactured in the machine shop. Usages of nomenclature and methods of designating various elements differ to such an extent that additional information must often be secured before work can be done, or misunderstandings of what is required result in wastes of materials and labor. Adoption of definite and simple standards will eliminate all such conditions, promote efficiency, and prevent wastes.

Much work has been done looking towards the standardization in the machine-shop industries, yet there remains a field for further standardization so large that by comparison the surface has hardly been scratched.

The American Engineering Standards Committee has, for several years, carried on much standardization work of an important and highly commendable nature. This body, in collaboration with various engineering organizations and manufacturers, has already completed several projects of standardization and simplification whose value already can be computed in dollars and cents. With a standardization body available that is ready and willing to coöperate and to act as a clearing house for further projects, it only remains for those engaged in the machine-shop industries to bring forward these projects and to secure coöperation among themselves. Even in the latter respect the A.E.S.C. can be of great value because it has already established connections with many important firms and industries.

With the purpose of acquainting those not already familiar with current standardization work relating to the machine-shop industries, a list of projects carried on by the A.E.S.C. is given below. In each instance the present status of the project is indicated together with explanatory comments. It is believed that general interest in the subject can thus be stimulated and that more people will realize the advantages of standardization and so interest themselves in its promotion.

A number of safety codes are also included in the list below because these are in reality the standardization of working conditions and of safety devices for the purpose of eliminating human wastes in industry.

Completed Standards

1 American Standard Screw Threads for Bolts, Machine Screws, Nuts, and Commercially Tapped Holes. This work was carried on for several years by a most able committee, and resulted in the adoption of definite standards approved by the A.E.S.C. in May, 1924. The standard includes the following sub-headings: Terminology, Form of Thread, Thread Series, Classification of Fits, Tolerances, Coarse-Thread Series, Fine-Thread Series and Appendices, covering Derivation of Tolerances, Relation of Lead and Angle Errors to Pitch, Diameter Tolerances, and Tolerances for Loose Fits.

2 Safety Code for the Use, Care, and Protection of Abrasive Wheels. Code adopted in 1922. Recent experimental work on higher-speed wheels has been carried on looking towards the revision of the 1922 code.

3 Safety Code for Power Presses and Foot and Hand Presses. Code has been constructed. A revision is projected when the sectional committee again convenes.

4 Safety Code for Mechanical Power-Transmission Apparatus. This Code was adopted by the A.E.S.C. in July, 1923.

5 Diameters and Lengths of Cold-Finished Shafting. Approved as a tentative American Standard in December, 1924.

6 Screw and Flat Stock Keys. Approved as tentative American Standard in June, 1925.

7 Safety Code for the Protection of Head and Eyes of Industrial Workers. Issued as Bureau of Standards Handbook. Second edition dated December 29, 1922.

In addition the A.E.S.C. and the Department of Commerce have lent valuable aid to trade associations and industries in completing the elimination of obsolete and superfluous sizes of manufactured articles. One example of this work is the elimination of many sizes of small tools which had become obsolete or unnecessary.

Projects in Process

1 Screw-Thread Gages and Gaging of Screw Threads. One portion of this work has already been completed. The section on screw-thread gages and the gaging of screw threads has already proceeded to the point where a draft proposal is under consideration by the committee responsible for the project.

2 Ball Bearings. A draft proposal of this standard is already under consideration by the committee.

3 Tolerances and Allowances for Machined Fits in Interchangeable Manufacture. This standard has been practically completed but is being held up for minor additions to be approved by the A.E.S.C. The final standard should be ready by the end of the year.

4 Methods of Gaging.

5 Specifications for Gages. Drafts of these standards are now before the committee for approval.

6 Small Tools and Machine-Tool Elements. The paper by Mr. Einstein printed in the August issue of MECHANICAL ENGINEERING deals most ably with this subject.

7 Gears. This project has been divided into eight subdivisions, due to its broad scope, and at least one of the sub-committees has already reported. Much of this work is yet in its preliminary stages.

8 Safety Code for Machine Tools. Project just started.

9 Formulas for Use in Determining the Size of Transmission Shafting. Two sections of this standard have been completed. The third section is now before the committee for final approval.

10 Bolt, Nut, and Rivet Proportions. This project is divided into six sub-divisions and is progressing favorably. An early issue of these important standards is expected.

11 Safety Code for Forging and Hot-Metal Stamping. A draft of this standard is practically complete.

12 Safety Code for Plate and Sheet-Metal Working. Combined with 11 above.

13 Pins and Washers (Machine Elements). Project just started.

14 Transmission Chains and Sprockets. Project now under way. Committee announced in A.S.M.E. *News*, January 22, 1925.

15 Scientific and Engineering Symbols and Abbreviations. This project is just getting under way.

Material Specifications

The A.E.S.C. has adopted the specifications issued by the American Society for Testing Materials covering carbon- and alloy-steel forgings, Bessemer screw stock, and open-hearth screw stock. Another project is the numbering of steels. So far this excludes tool steels, but it is hoped that this standard will be extended to include the latter items.

Projects Contemplated

1 A safety code for cranes, derricks and hoists is under consideration by the Safety Code Correlating Committee.

[1] Factory Superintendent, Chief Engineer, National Twist Drill & Tool Co. Mem. A.S.M.E.

2 A Standard for drawings and drafting-room practices has been discussed, and was given some publicity in the August issue of Mechanical Engineering. The early adoption of this standard will make a very valuable addition to the already imposing list. Much confusion exists at the present time as well as different usages which have been developed by various industries.

3 Standardizing graphical presentations is another important project under consideration. The A.E.S.C. by request of the A.S.M.E. has authorized the appointment of a committee to deal with this important subject in detail.

No attempt is made in this article to list all the various A.E.S.C. standards, relating to other than the machine-shop industries. The 1925 Year Book issued by this committee lists approximately 170 distinct standardization projects in various stages of completion.

It will be realized from this that the standardization work is progressing rapidly. The continuance of the work and the consideration of new projects depends largely on the coöperation of engineers and industries that are interested.

The machine-shop industries should not fail to avail themselves of the opportunity to establish broad and uniform standards for the multitude of things used and manufactured. If a sufficient demand is indicated for standards, both the A.S.M.E. and the A.E.S.C. are ready to give them consideration and publicity so that all interested parties can have a voice in the final drafts.

Standardization is certainly becoming one of the important keys to progress. Let us use it.

Logging—Newsprint Manufacture—Hog Fuel

Discussion of Papers on these Subjects Presented at the Portland Regional Meeting, Portland, Ore., June 22-25, 1925

At the technical session of the Portland (Ore.) Regional Meeting of the A.S.M.E., held in the Assembly Hall of the Hotel Portland, June 22, 1925, W. F. Durand, President of the Society, was introduced by William C. Shaw, Chairman of the Oregon Section, and presided at both morning and afternoon sessions. Papers were read by C. W. Morden on Technical Features of High-Speed Newsprint Manufacture;[1] and by E. C. Hutchinson on The Oak Grove High-Head-Turbine Development of the Portland Electric Power Company.[2] At the afternoon session the following papers were read: The Port of Portland's 30-In. Diesel-Electric Pipe-Line Dredge *Clackamas*,[3] by James H. Polhemus; Steam Logging,[4] by Joseph W. Gill; Electric Logging,[5] by P. A. Wickes; The Overhead-Cableway Method of Logging,[6] by Spencer Miller; and The Utilization of Wood Waste as Fuel in Steam Power Plants,[7] a symposium arranged for and coördinated by H. S. Bastian; including Combustion of Wood Waste from Lumber-Manufacturing Plants, by H. W. Beecher; Boiler-Room Operation of Wood-Refuse-Fired Steam Plants, by Claire C. Simeral; Settings for Hog-Fuel-Burning Boilers, by C. L. Young; and Hog-Fuel Conveyors, by A. C. Sullivan.

An account of the discussion of these papers follows:

High-Speed Newsprint Manufacture

In the discussion of Mr. Hutchinson's paper, in answer to questions relative to the voltage and the length of the transmission lines, it was said that the present operating voltage was 60,000 volts, but that the line was designed for an ultimate voltage of 110,000 volts, the ultimate length of the transmission line being 50 miles. The author explained that the Moody spreading draft-tube increased the efficiency of the turbine at partial loads but that with high-head turbines at full load there was little difference in efficiency due to its use. He said that the Oak Grove turbine developed an efficiency of 91 per cent or better at maximum loads, and at partial loads the efficiency had been much higher than had been anticipated. The Moody draft tube also decreased vibration in high-head turbines. The cone at the center of the draft tube had to be designed for a given set of conditions.

R. L. Daugherty[8] pointed out that the ordinary type of draft tube was satisfactory under conditions where a streamline flow parallel to the axis of rotation obtained, which was approximately the condition at full load with low-specific-speed types of runners, but in high-specific-speed turbines and with low-specific-speed turbines at anything besides full load, there was a rotational flow which caused an unstable condition at the center, a condition which could be removed by filling up the central portion with the cone.

The Overhead-Cableway Method of Logging

In discussing Mr. Miller's paper on this subject, J. A. Kinkead[1] asked about the difference in the method described in the paper and the one used at Elk, California, by the Goodyear Redwood Lumber Co., covering a distance of more than 5000 ft.

J. M. Meany,[2] who presented the paper in the absence of Mr. Miller, replied that in the case referred to the effective distance which the lumber was yarded was not five thousand feet although the span was of that length, because the ends of the span were on the tops of two mountains and the railroad ran between them.

Utilization of Wood Waste

A. A. Straub[3] contributed a written discussion of the papers dealing with the utilization of wood waste as fuel in steam power plants. Analysis of conditions prevailing in hog-fuel furnaces, he wrote, when this fuel was fired on flat grates, proved the author's statement that there was a decided tendency to stratification of gases from the fuel bed. With this fact in mind it was apparent that the addition of refractory to a furnace in a manner to improve ignition of the fuel, ignition and mixing of the gases of combustion, should be of benefit.

During the process of Mr. Straub's recent studies incident to improving conditions in the burning of this fuel, it had been manifest that investigations would need be carried out along the lines noted in order to secure proper combustion and decrease the manual labor and high degree of skill required on the part of the firemen in order to secure results when operating the furnace in question.

The particular furnace under discussion had 12 by 12 ft. flat grates with a flat arch 6 ft. above grates and a drop nose. The furnace was attached to a 600-hp. Stirling water-tube boiler. The furnace had no party wall and was fired through four holes in the top of the flat arch, the fire holes being equipped with balanced dampers. The fuel existed on the grates in four overlapping cone-shaped piles, the center of each at about the center of one-fourth the grate area. In order to improve conditions of combustion the face of the bridgewall was made vertical and a pier and wing walls had been added. The bridgewall had had to be decreased in height and the pier not built entirely up to the arch in order to allow sufficient area through the throat. The addition of this refractory had resulted in improved and quicker ignition of fuel in the furnace. It had permitted the use of wet fuel, which had been stored a considerable time, without excessive loss of evaporative capacity. It had aided in mixing and combustion of gases, in that the lean and

[1] See Mechanical Engineering, June, 1925, pp. 495–496.
[2] See Mechanical Engineering, June, 1925, pp. 449–454.
[3] See Mechanical Engineering, August, 1925, pp. 605–612.
[4] To appear in October issue of Mechanical Engineering.
[5] To appear in October issue of Mechanical Engineering.
[6] See Mechanical Engineering, July 1925, pp. 527–534.
[7] See Mechanical Engineering, July, 1925, pp. 545–554.
[8] Professor of Mechanical and Hydraulic Engineering, California Institute of Technology, Pasadena, Cal. Mem. A.S.M.E.

[1] Sales Representative, Parkersburg Iron Co., Rialto Bldg., San Francisco, Cal. Mem. A.S.M.E.
[2] Lidgerwood Mfg. Co., 101 First St., Portland, Ore. Mem. A.S.M.E.
[3] Betterment Dept., Electric Bond & Share Co., 71 Broadway, New York, N. Y. Mem. A.S.M.E.

rich gases had been brought into contact in a zone of incandescent refractory surface.

This additional refractory had resulted in increased boiler evaporating capacity from 145 to 190 per cent builders' rating with same intensity of draft available at the boiler damper. The degree of care the firemen had been compelled to exercise and the actual manual work performed had been noticeably decreased.

While it had not been feasible in the time allotted to study the variation in gas composition across the setting, visual observations through a side observation door back of the bridgewall, had indicated good mixing and noticeable decrease in stratification after the installation of the additional refractory. The intermittent method of firing had been employed on all tests, as fuel-handling equipment did not readily lend itself to the continuous method of firing.

Hogged fir had been used in all tests and the percentage of CO_2 in the flue gas had been found to depend considerably on the character of the fuel fed and the steaming rate. When operating at 150 per cent rating with reasonable care in firing no difficulty had been encountered in maintaining 14 to 16 per cent CO_2 without the presence of CO.

Mr. Straub had felt that no furnace brickwork trouble should be encountered and had estimated temperatures prevailing of the degree stated by the authors. When using oil as reserve fuel, which was the usual practice in this section, there was a danger; and it had actually occurred in practice, that the checkerwork at the oil burners had fused over quickly when placing burners in operation without proper precaution and preparation. This was due, as the authors pointed out, to the fluxing action of the wood ash on the firebrick. As the boilers were usually fired from the rear with oil, the wood ash from front-firing with hogged fuel was deposited on the checkerwork, at the oil burners. In order to overcome this difficulty at the plant where Mr. Straub had had the unit under study, two plans were being tried. One plan consisted in passing a steam lance under the checkerwork before placing oil burners in service, thus dislodging the ash and permitting the draft to carry it away. The other plan consisted in laying up the checkerwork with feather-edge brick to prevent accumulation on the checkerwork, but also still using the steam lance.

In order that the results of studies might be made applicable to every-day operation, the following firemen's hints had been written, and made available to the firemen.

Hints for Firing Hog Fuel

1 Keep pit doors open at all times on boilers in service.
2 Control the steam pressure and main air supply with the boiler damper. Supply the necessary fuel to generate the quantity of steam required.
3 Admit air at all times over the fire by keeping the fire doors partly open. Regulate the opening of the fire doors to prevent the discharge of dark smoke from the stack. The stack should be clear or a light haze should leave it. Planer shavings and dry fuel require wider fire-door opening than wet, fine fuel.
4 Fire often and in small quantities rather than in large quantities and less frequently.
5 Keep the fire thin in front and heavy in rear; top of rear fuel cones should be closer to arch than top of front fuel cones.
6 Fire in front should reach about four to six inches over the dead plate at the center of firing doors. It is better to have the grates slightly uncovered at the front than to have too much fuel charged.
7 The fire at the front should always appear bright and active. The fire-door opening should show bright when viewed from the firing floor, a dark door shows the fuel bed too heavy.
8 The fire should be heavy enough in the rear to prevent vicious flame coming through at the bridgewall.
9 With coarse, dry fuel the fire should slope at rear to slightly below the top of the bridgewall. With wet fuel it should be somewhat thinner. With fine, wet fuel the fire at the rear should slope to about half-way up the bridgewall.
10 Fire in the rear should be carried heavier for coarse and dry fuels and thinner for fine, wet fuels.
11 At no time entirely close the ashpit or firing doors on a boiler in service; use the boiler damper for main control of air supply.

These hints practically conformed to statements by the authors with adaption to the particular equipment under study.

Mr. Straub wrote that his findings corroborated the statement that it was advisable to admit all air for combustion through the front doors, both ashpit and fire doors, and through no other openings in the furnace. It was the practice at the plant where investigations had been made to keep the ashpit doors wide open at all times on boilers in use and vary the fire-door opening as called for by firing hints. In order to facilitate the opening and closing of doors by the firemen from firing platform, fire-door-control extension handles had been installed.

The following analysis of a sample of pit refuse which Mr. Straub had prepared and analyzed would probably be of interest.

Kind of fuel	Hogged fir
Average boiler evaporation rate, per cent builders' rating	125
Length of run represented by sample, hr	24
Analysis of pit refuse, per cent	
Combustible	5.22
Non-combustible	94.78
Total	100.00

Representing a loss of 1/2 of 1 per cent of fuel to the ashpit when using hogged fir as fuel.

The hog-fuel conveying system presented a problem for considerable study. Due to the lack of this fuel to follow any tendency other than to obstruct flow and its ability to arch over, too much care could not be exercised in designing and installing such a conveying system.

Claire C. Simeral, in answer to a question, said that hemlock was fired the same as fir, only a little more draft was used and less fuel was kept in the furnace.

H. W. Beecher said that the unit 200 cu. ft. was used instead of a cord, 128 cu. ft., because it was a more convenient figure to handle, and because at the time that the tests had been made to which he had referred, the company had been buying wood as cordwood and also buying wood as hog fuel. When they had tried to determine how many cubic feet of hog fuel would be equivalent to a cord of slabs, the average was 177 cu. ft. which, for convenience, was made 200 cu. ft., a correction factor being applied.

Lee Straub[1] spoke of doing considerable work in the last few years in connection with hog fuel. Lately at a nearby city blue water-gas had been produced from hog fuel. Determinations indicated, he said, that it was possible to produce from 20,000 to 25,000 cu. ft. of blue water-gas, of about 300 B.t.u. per cu. ft. heating value, from a unit of hog fuel. Public-service commissions had not yet approved, in many locations, of the use of 300-B.t.u. gas for domestic consumption, so in order to meet present regulations this gas had to be carbureted and was known as carbureted water-gas. From a unit of hog fuel, and by using 2.5 gal. of fuel oil per 1000 ft. of gas produced, 550-B.t.u. gas could be produced. In experiments at Grants Pass the fuel cost of this gas had been about 16 cents per 1000 cu. ft. as compared to the use of 11 gal. of fuel oil at 6 cents per gal., or 66 cents per 1000 when using oil for that same purpose.

It was his belief that even better results might be obtained in boiler practice if the hog fuel were fired in a hot-gas producer and the producer gas carried over a bridgewall with air admitted in the same manner as in the large steel plants for the production of steam from blast-furnace and producer gas, higher furnace efficiency being obtained thereby.

O. F. Stafford,[2] said that not only should all of this wood waste possible be used in the production of energy, but all of it should be salvaged that might be possible in the production of the most valuable chemical product obtainable from wood, namely, paper and pulp. There was enough waste material suitable for paper and pulp to supply an enormous percentage of the world's needs. The question of its utilization was a question hedged about upon the one side by economic problems and upon the other side by technical ones. The technical problems were in a fair way of solution. The economic problems of course must be solved very carefully. That they were at the beginning of a solution in this territory, he had no doubt.

Another way in which this wood might be utilized economically was by converting it into products obtainable when wood was subjected to the action of heat. The production of charcoal from wood was an old art. It had within the last one hundred years been found possible to obtain certain other useful products from wood; particularly wood alcohol. Besides wood alcohol a considerable amount of acetic acid was obtained by the destructive

[1] San Francisco, Cal.
[2] Dean of Chemistry, University of Oregon, Eugene, Ore.

distillation of wood. Charcoal, acetic acid, and alcohol had been three important things which had been obtained in that way. The world needed a considerable amount of these things; it could use possibly more than it was getting.

In addition to these three things there were some others which were of use, such as wood oils. The pitch obtained from the destructive distillation of wood was coming into its own these days, particularly as the rubber manufacturer in making rubber tires had found it a substitute for more expensive carbon compounds used in the compounding of rubber. There was also the prospect of gas production—all of these things being possible as wood was subjected to the process of destructive distillation.

In this region undoubtedly the most valuable product would be charcoal. It had been found feasible to convert the charcoal obtained in this manner into a good fuel briquet. This was being done now upon a large scale in at least two places in the East, and the problem could be handled on the Coast just as well. This charcoal would represent an enormous potential resource for that part of the world, in case it was found possible to go ahead and produce it.

The practice had been in the past in this country for each man who needed a forest product to go into the forest and hew his own path, taking out the things which he needed and discarding everything else. This had meant a tremendous loss, because in no industry was it possible to do this without discarding a great deal of the material which would be useful for something else. In some of the European countries, in the Scandinavian countries particularly, the understanding between the lumber manufacturers and the paper manufacturers and the wood distillers was such that they worked absolutely together. In Scandinavia, forest products were segregated into those best suited for structural purposes and those which were useful for making pulp, and the residues were used for fuel or for wood-distillation purposes. In this country we should have to go far in order to get such a correlation, but which should be striven for, and until we did measurably attain it we should have no adequate solution of our wood-waste problem.

N. E. Ayer[1] said that he had devised a machine for removing the bark from the log so as to make handling at the sawmill easier, and to reduce cost in hauling logs, the bark being a big factor in the freight charges. The bark averaged about 15 per cent of the log, both in bulk and in weight. But he had been unable to find out what to do with this bark, and had written to the Forest Products Laboratory in Madison, Wis., without obtaining satisfactory information. Did anyone know of a commercial use for the bark? When dry, the bark was very easily pulverized, and he thought that there might be some great commercial use for it, such as using it as a fuel.

Professor Stafford said that for making gas, bark had a value which was comparable to that of wood itself. It had a high fuel value and could be used, if properly prepared, as fuel, just as hog fuel was used. There were some kinds of bark, however, for which perhaps a better use could be found. The question as to whether barks rich in tannin could be segregated advantageously, and thereby made the raw material for an extraction industry, was one on which he had no figures. Douglas fir bark, he believed, had been found by some experimenter to yield a product very similar to pulverized cork.

A. N. Walstad[2] told of the use of saw-mill waste by the City of Minneapolis as long ago as 1890, and made a plea for the elimination of waste in the lumbering industry.

H. W. Beecher offered an example to illustrate the effect of high cost of handling on the cost of hog fuel. There had been a number of experimental plants put in for briquetting hog fuel, he said, but they were still in the experimental stage.

C. W. Morden called attention to the hearth type of furnace. So far as he knew this type of furnace had not been used around Portland for burning hog fuel, except at the plants of the Crown-Willamette Paper Company, where it had been used for a number of years, the results having been such as to indicate that it might have an application on larger boilers than were in use at that plant. He thought that this type of furnace would be given greater consideration in the very near future.

[1] President, St. Johns Lumber Co., Portland, Ore.
[2] Walstad Machine Co., Tacoma, Wash. Mem. A.S.M.E.

William S. Hill[1] said that a good many years ago his company had used, as it was still using, electric cars holding about seven units to transport hog fuel from one of the mills to their plant. They had not been able to get enough fuel from that one plant, so they had recently put on two five-ton trucks that carried three units of fuel to each truck. This had enabled them to cut their cost of transportation down to about 50 cents a unit, and it had helped out one mill in particular. This mill had just been ready to go to the expense of putting in a large burner to take care of the waste so of course they had got a favorable contract for taking the fuel away from that particular mill. There were other mills and smaller plants and industries which had wanted this company to carry the fuel away for nothing, because it was a fire hazard around their individual plants, and they must pay out money for fire insurance which they would not otherwise have to pay in case the fuel were taken away.

The German Museum of Applied Science

THE OPENING in Munich, Germany, on May 7, of the Deutsches Museum von Meisterwerken der Naturwissenschaft und Technik marked the culmination of twenty years' efforts and was an occasion of national importance. This museum was conceived and brought into being under discouraging conditions which were aggravated by a war and a revolution, Dr. Ing. Oskar von Miller, the well-known consulting electrical engineer, having had the foresight and wisdom to bring the project to completion. The opening was coincident with Dr. von Miller's seventieth birthday and was accompanied by an allegorical procession on May 5; and the actual opening was marked by a play, specially written by Germany's foremost poet, Gerhard Hauptmann. Although the museum is not a state affair, great interest has been taken in it by governmental authorities, even to the extent of an appropriation by the Reichstag of about $25,000 so that a selected number of young people may spend a week in Munich to visit the collections.

The building, set on an island in the river Isar, is 345 by 325 ft., and 80 ft. high, and has a tower 210 ft. high. The floor area is 350,000 sq. ft. The collections are arranged systematically, an introduction through the economic aspects of geology leading to life-size models of mining operations, of coal, salt, and ore, the methods and machinery involved being thoroughly illustrated as well as the occurrence of the ores. In a similar manner, other branches of engineering are illustrated by models, by typical rooms of smiths, chemists, paper makers, millers, etc.; by famous and epoch-making inventions, comparisons of primitive and modern machinery and tools; and by pictures of ancient arts and industries. The construction of roads, railways, tunnels, canals, locks, and subways is also illustrated, as well as types of prime movers; methods of transportation, vehicular, railway, marine, submarine, and aerial; and the electrical arts and industries. The sciences are not forgotten, nor are standards and methods of measurement. A hall of honor contains paintings and busts of famous men, and less famous men are remembered in smaller halls.

An interesting exhibit worked out by Dr. Bauersfeld of the firm of Carl Zeiss, of Jena, is a Ptolemaic planetarium, in which images of the fixed stars are projected on the dome of a darkened room. The mechanism, set in motion, gives in four minutes a representation of the daily movement of the heavens. Similarly the yearly courses of the sun, moon, and planets can be shown in about five minutes. So popular has this astounding display become that ten similar devices have been ordered for other cities in Germany.

Not the least remarkable feature of the museum is the popularity which the project has acquired throughout Germany and the interest in engineering, science, and industry which it has stimulated. One reason advanced for the great popular enthusiasm is that since the War there has arisen in Germany a conviction that she must look to technical progress to meet Reparation demands and to reëstablish her lost position.

[1] General Superintendent, Power Production and Electric Railway, Grays Harbor Railway & Light Co., Aberdeen, Wash. Assoc-Mem. A.S.M.E.

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and Attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

The American Society of Mechanical Engineers

29 West 39th Street, New York

The Opportunity of the A.S.M.E. Machine-Shop Practice Division

THE EXCELLENT attendance and the stimulating addresses that featured the New Haven Machine Tool Exhibition last year and the interesting topics and pertinent discussion scheduled for the sessions at the coming exhibit are gratifying evidences that the members of The American Society of Mechanical Engineers are becoming increasingly aware of the valuable opportunity that the Society offers for the consideration of the technical problems underlying the operation of American machine-shop industries. The comment has often been heard that the A.S.M.E. is primarily a power Society, and a study of recent Transactions may seem to confirm this opinion, but the fact is that the larger part of the membership is engaged in the engineering and executive work of the many manufacturing industries of the country. Obviously, the problems of the gigantic steam plant lend themselves to accurate technical and economic analysis and recent advances in the power field have been so revolutionary and extensive that it is natural for the Society to give ear to the power engineer, for the activities of the Society reflect the engineering movements of the day. Look over the list of past officers and you will find that the leaders of progressive movements in engineering become the active leaders in the Society. Now with the problem of producing increasing volumes of power at decreasing costs we have the companion problem of producing what the world needs in manufactured goods at lower and lower costs with no sacrifice of quality. It is in this field that the Machine-Shop Practice Division has its opportunity.

In cultivating its field, the Division has set the task of encouraging original research, of fostering education by persuading educational institutions to devote more attention to the unsolved problems of machine-shop practice, of advancing standardization, and of promoting the exchange of experience in machine-shop matters between members of the Society and with other societies. Today machine-shop practice is but little more than a collection of conventions and traditional practices and with minor exceptions few engineering data are available. The field of the Division is large, its problems are diverse, but it offers the widest opportunity for development along scientific lines of any field of mechanical engineering. The duty of the Division is not to record past performances but to stimulate thought and achievement.

It is fortunate that the Machine-Shop Practice Division meets once a year in the congenial atmosphere of the machine-tool exhibit at New Haven. The auspices of Yale University, the New Haven Chamber of Commerce, and the New Haven Section of the A.S.M.E. insure a setting that promises well for the future of the work of the Division. In view of the work to be done and the service to be rendered the increasing participation in the sessions by members of the Society is most gratifying.

Walter F. Dixon.[1]

Machine Tools and Production Costs

THERE is convincing evidence that the manufacturers of this country are not utilizing to the full the opportunity the machine-tool builders afford for reducing manufacturing costs on production operations. New machines embodying the experience and skill gained in solving wartime production problems and in the development of the automotive industry are available but the demand for them does not appear to be equal to the need for them. Machine builders flourish only when industry is expanding rapidly and as they must now depend on "wear-out" and obsolescence for their business, they are presenting tools of greater producing power to accelerate the obsolescence rate. They are waging a campaign of education in the performance possibilities of the newer machine tools and they accept the burden of proving that a tool will pay for itself in a stated time. All of this is highly stimulating to industry but there is inertia to be overcome and time is needed for the campaign to permeate the nooks and crannies of the machine shop and search out the relics of past machining generations.

Recently, leading industrial engineers have shifted the emphasis once placed upon organization technique to equipment and equipment maintenance. They agree that obsolescence must be something more than an accounting gesture. They state that the manufacturer must not be satisfied merely to have his machines running 100 per cent of the time. He must be sure that they are producing better and more cheaply than those of his competitor or he should replace them. He should know that up-to-date manufacturers crowd their machines to a "reasonable maximum capacity" because of the rapidity with which machinery becomes obsolete. Some users estimate five years as the life of a production machine tool and use it accordingly. Constant scrutiny of the factors making up the machine rate for each tool and comparison with probable rates for new equipment are the essentials of success in getting the best results from all types of machinery.

The leaders in the producing industries appreciate all this but there are many following a restrictive policy which, applied to machine equipment, leads to disastrous results. Witness the expenditures of many times the value of a machine for repairs because of an edict of "no capital expenditure for machine tools." And the lessons apply not only to mass-production operations but have some influence on all machining jobs. New machines are being developed daily that will produce more cheaply than machines now in use. The progressive manufacturer will scan the field for them, but he will not buy on specification at so much per pound as he does cotton waste. He will demand performance and reduced production costs.

The efforts being made by machine-tool builders to embody their skill and experience in the modern machine tool and the searches for more economical metal-working equipment by up-to-date manufacturers naturally lead to a clearing house for the latest information. This is at hand in the New Haven Machine Tool Exhibition, which is held annually at the Mason Mechanical Laboratory of the Sheffield Scientific School, and which is being conducted according to a policy and under auspices that insure the display of machine tools embodying the newest developments. The Exhibition is unique in that it is carried on under the direction of representatives of two great educational organizations, Yale University and the A.S.M.E., and a commercial group, the New Haven Chamber of Commerce, and it is fulfilling a valuable purpose.

Originally conceived as a part of the training in Mechanical Engineering at Yale, the exhibition has now broadened into a tremendous influence for the advance of all metal-working industries.

[1] Chairman, A.S.M.E. Machine Shop Practice Division, Works Manager, Singer Mfg. Co. Mem. A.S.M.E.

Ball Indentation Tests

Hardness and hardness measurement have had constant attention for many years and there are several satisfactory devices for comparative measurement of each kind of hardness. As the terms in which each hardness is measured are not intertranslatable, as hardness and machinability are not accurately defined qualities, and as a hardness factor may not be transformed into a measure of machinability, the state of the art of measuring these qualities in absolute terms is still very much where it was twenty years ago. In this issue of Mechanical Engineering, Mr. Ault presents an interesting comparison of many studies made in the use of the various types of ball-indentation methods of measuring one type of hardness. His conclusion is of distinct interest for he points out that it would be more accurate to measure the depth of the indentation than to follow the present method of measuring the diameter of the impression of the ball. In this he agrees with Mr. DeLeeuw, who, at a meeting of the Research Committee on Cutting and Forming of Metals, pointed out that absolute values cannot be obtained by keeping the impressing force constant and measuring the diameter of indentation. Mr. DeLeeuw advocated the measurement of the force required to make an indentation of a standard depth using a standard ball. This method is practical and simple and would be of outstanding interest to the Committee on Cutting and Forming Metals as it would make available to the shop man the results of research in cutting practice. The present Brinell method is essentially an inspection method and, as such, its results are not exactly comparable outside of any given shop or process.

It should also be clearly borne in mind that the Brinell test shows the *average* quality over the areas of the indentation. This may not be of importance in a material of uniform texture such as rolled steel, but may be misleading where small hard particles are embedded in the soft matrix, as, for example, in the case of phosphor bronze. This may perhaps throw light on the reason why there does not seem to be any clear relation between either Brinell hardness and machinability or Brinell hardness and the ability of the material to act as a bearing material.

Progress under Compulsion

Inventors and manufacturers have at all times been more or less at loggerheads. Some ten years ago a gentleman connected with the automobile industry publicly stated that the best thing for his particular industry would be to have the U. S Patent Office shut down for about five years. Of course this was out of the question, yet the automobile has been tremendously improved since that time, to the vast benefit of every one concerned.

In general, however, industry does not take up new inventions or processes until it is absolutely compelled to do so. An instance of this came up only recently and attracted a good deal of popular attention. It was announced that German chemists had developed a process for synthetically producing methanol (wood alcohol) at a cost which threatened the entire American wood-alcohol industry whose investment is estimated at $100,000,000. The saddest part of the story, however, lies in the fact that the basic patents on this process were issued in the United States some ten years ago, were seized by the Alien Property Custodian in 1917, and have since been practically available to any American concern that wanted to use them. It seems, however, that under the methods described in the patent the cost of production was such that the synthetic product could not compete with the product of distillation of hard wood. More research was required, but no one in America, notwithstanding that an industry of $100,000,000 was at stake, cared about it sufficiently to push the investigation through to the last step—that of developing a commercial process out of a laboratory process.

Once in a while, however, progress under compulsion is a blessed thing. As an instance, only ten years ago the Diesel engine entered the marine field, but so formidable has its competition become that steam navigation bids fair to be practically annihilated unless something radical is done before long to rehabilitate its standing. Even in such enormous vessels as the recently built passenger ship *Aorangi* the Diesel engine seems to give excellent efficiency.

Now, in the face of such an enemy hammering at the gates of the Empire of Steam, every one within this latter domain seems to have awakened from a long and comfortable slumber. Sir Charles Parsons, who advocated higher steam pressures for a long time to a world that would not listen, has suddenly become an honored prophet in his own country, as would appear from a paper read at the Spring Meeting of the Institution of Naval Architects by Sir John Biles. Yarrow & Company have designed a boiler for 500 lb. pressure and 700-deg. superheat for marine engines. James Howden & Company are building a boiler to employ 315 lb. pressure and 700-deg. superheat, and other firms in England are working along the same lines. Any one who would have suggested such pressures and superheats for marine use three years ago would, in the eyes of marine engineers both here and across the water, have been a fit subject for a mental test, as would probably have also been a man in America at the same time who claimed that the hardwood-distillation industry was threatened by synthetic products.

That compulsion can result in progress has been known for a long time. Whether it is a worth-while policy for an industry to begin research only when its very existence is threatened by progress elsewhere is, as Kipling says, "another story."

Steam-Nozzles Research

The appearance of the Fourth Report of the Steam-Nozzles Research Committee of the Institution of Mechanical Engineers again stimulates discussion of the shape of the efficiency curve with its widely separated maximums at about 300 and 1800 ft. velocity. This is particularly interesting since so many of the European manufacturers have taken up the building of machines embodying these general principles and even American manufacturers in certain designs have shown similar tendencies. The Loesel designs have made further progress but no satisfactory tests have been reported. In general it may be said that the many-stage small-diameter high-pressure end has taken a prominent place in current design while all the larger machines show the higher maximum in use at the low-pressure end. Most of the discussion has centered on the advantages of the finishing of nozzles all over, while there has been some published criticism of the trend of the nozzle-efficiency curve at low velocities. Considerable stress has been laid on the thickness of lands at the exit end of the nozzle together with the "flapping" action observed in the tests, the cause of which has not been developed. The criticism from the designers using the reaction type that the tests confirm reaction-turbine practice has been touched on in the discussion, and the committee has been urged to continue the tests with nozzles of the reaction type.

Similar investigations which have been carried on in this country but not published are reported to confirm the English results and in general leave the impression that we may expect higher overall efficiencies in steam-turbine performance of the later machines.

A Survey of Air Transportation

The exhaustive survey of air transportation, voted by the American Engineering Council after having received the report of its Committee on Aircraft (See Mechanical Engineering June, 1925, p. 518), becomes possible through the appointment by Secretary Hoover, of the Department of Commerce, of a committee of six to undertake the study, thus securing for the Council, governmental assistance and backing. J. Walter Drake, Assistant Secretary of Commerce, will direct the work as chairman. A field staff is being organized with Professor Joseph W. Roe, head of the department of industrial engineering, New York University, and chairman of the Councils' Committee which submitted the preliminary report, as Director and Vice-Chairman of the committee of six.

The other numbers of the committee are: Dr. W. F. Durand, President of The American Society of Mechanical Engineers; Prof. E. P. Warner, head of the aeronautical-engineering course at the Massachusetts Institute of Technology; Luther K. Bell of Washington, traffic manager of the United States Air Mail; C. T. Ludington of Philadelphia, aircraft operator.

The field staff will include J. Parker Van Zandt of the United States Air Service, who has recently made an extensive study of civil aviation abroad, and Prof. Alexander Klemin, head of the aeronautical engineering course in New York University. The staff, of which Lieutenant Van Zandt is secretary, is already at work and expects to complete its task in about six months.

The coöperation of the Bureau of Standards and of governmental agencies of the Department of Commerce who are studying commercial aviation abroad will greatly facilitate the work of the committee.

"At the outset," said Professor Roe, "a comprehensive survey will be made of commercial air lines in other countries, including services offered, volume of traffic, safety, regularity, financial status and equipment, cost of operation, government aid and maps showing routes, landing fields, distances, and densities of traffic.

"This will be followed by a similar study of what has been done in the United States, where most of the experience is confined to the air mail. There will be a study of the geographic, economic, and operating conditions in the United States, the volume of business possible for mail, express, and passengers, the possible routes and their relation to existing forms of transportation.

"There will be a review of the government aid offered to water transportation, such as the Coast Survey, lighthouse service, harbor improvements, etc. and how far and in what ways similar aid can be given commercial air transport. There will be a study of the applications of aircraft to industrial uses other than transportation, such as combating the boll weevil, forestry patrol, aerial photography, surveying, etc.

"If commercial aviation is to be developed in this country, it will probably be largely through private financial support rather than direct governmental aid, as in Europe. This support must come from investors and banks, who will not and cannot be interested in this field until the facts relating to the commercial possibilities, risks, etc. are available."

Machine-Shop Sessions at New Haven

SIMULTANEOUSLY with the Fifth National Machine Tool Exhibition, which will be held in Mason Laboratory of Mechanical Engineering, Yale University, September 8 through 11, 1925, there will be a series of technical sessions conducted under the auspices of the Machine-Shop Practice Division of the A.S.M.E. These sessions will be held in Dunham Laboratory of Yale University.

The program has been arranged as follows:

Wednesday Afternoon, September 9, 3.30 p.m.

Paper on Centerless Grinding, by W. J. Peets.

Wednesday Evening, September 9, 8 p.m.

Address by Mr. O. B. Iles, President of the National Machine Tool Builders' Association and of the International Machine Tool Co., Indianapolis, Ind., on The Future of the Machine-Tool Industry.

Address by W. H. Rastall, Chief of the Industrial Machinery Division Bureau of Foreign and Domestic Commerce, Department of Commerce, Washington, D. C., on The Foreign-Trade Outlook in the Machine-Tool Field.

Paper on All-Metal Airplanes, by William B. Stout, President, Stout Metal Airplane Co., Detroit, Mich.

Thursday Afternoon, September 10, 3.30 p.m.

Cylindrical Precision Lapping, by Paul M. Mueller.

High-Speed Cutting of Brass, by Luther D. Burlingame.

Thursday Evening, September 10, 6.30 p.m.

Round-Table Dinner Discussion. Four tables, subjects being: Inspection methods and precision measurements; power-press work; production milling; and shop training methods.

Friday Afternoon, September 11, 3.30 p.m.

Session on Shop Training Methods in Charge of Professor John T. Faig, Ohio Mechanics Institute, Cincinnati, Ohio. At this session addresses will be presented on different phases of the training problem by Mr. J. P. Kottcamp of the Pratt Institute, and by Mr. H. A. Frommelt of the Falk Corp., Milwaukee, Wis.

The papers by Messrs. Peets, Mueller, and Burlingame appear in this issue of MECHANICAL ENGINEERING.

The Machine Tool Exhibition is being conducted under the joint auspices of the Sheffield Scientific School of Yale University, the New Haven Section A.S.M.E., the Machine-Shop Practice Division of the A.S.M.E., and the New Haven Chamber of Commerce. During the previous four years the exhibits have been devoted particularly to new developments in the machine-tool field and have attracted large attendance.

Czech Engineers Celebrate 60th Anniversary

THE Sixtieth Anniversary of the Society of Technical Engineers of Czechoslovakia was celebrated at Prague on June 20 through 24, 1925. The festivities, at which President Mazaryk and his ministers were present, consisted of the laying of the corner stone of the Prague Technical School, the laying of the corner stone of the building for the engineering societies, and a visit to the Skoda Works. The A.S.M.E. was represented by Fred M. Oppermann of Holisov u Stoda, Czechoslovakia.

Summer Meeting of the A.S.C.E.

THE summer meeting of the A.S.C.E., held in Salt Lake City, Utah, July 8 to 10, was devoted chiefly to the subject of irrigation. The History and Problems of Irrigation Development in the West was the subject of an address by Dr. John A. Widstoe, Salt Lake City, formerly president of the University of Utah and of the Agricultural College of Utah, and member of the fact-finding commission, U. S. Bureau of Reclamation. A. P. Davis, formerly chief engineer of the U. S. Reclamation Service, spoke in defense of federal reclamation work.

R. E. Shepard of the North Side Twin Falls Land and Water Co., Jerome, Idaho, in speaking on the Financing of Irrigation Development, suggested the creation by Congress of a corporation fostered by the government somewhat as are the Federal Reserve Banks and the Federal Land Banks, to take over the operation of existing projects, to issue bonds for needed construction, and to look after reclamation projects in general.

Elwood Mead, Commissioner of Reclamation, spoke of the need for a revision in federal reclamation policy to meet new social and economic influences.

William Kelley, Colonel, Corps of Engineers, U. S. A., Chief Engineer, Federal Power Commission, Washington, D. C., read a paper on Coördination of Irrigation and Power. Irrigation and power, both being of importance in the Western States, require regulation of stream flow which may bring them into conflict. The paper gave a general presentation of the factors entering into the problem, illustrated the difficulties, and pointed out certain solutions by describing some of the specific cases which had come before the Federal Power Commission.

The Relation of Stream Regulation to Irrigation and Power Development was discussed by John C. Stevens, consulting hydraulic engineer, Portland, Ore., who pointed out that where there was a conflict between irrigation and power interests in the arid west, irrigation should have first claim.

S. T. Harding, associate professor of irrigation, University of California, read a report on the findings of the committee appointed to investigate the duty of water. The Consumptive Duty of Water was the title of a report by O. W. Israelson, professor of irrigation and drainage, Logan, Utah. M. C. Hinderlider, state engineer of Colorado, presented one view of interstate water problems and their solution, and Frank C. Emerson, state engineer, Wyoming, presented a widely contrasted view.

There were presented also papers on highway problems, on city planning, a symposium on working stresses in steel construction, and a symposium on reinforced concrete unit stresses and field control.

Building Construction and Earthquakes

COMPETENT observers of the destruction wrought by the Santa Barbara earthquake confirm deductions drawn from experiences at Tokio and San Francisco in the matter of the resistance of buildings to injury from such causes. Dr. Bailey Willis, president of the Seismological Society of America and professor emeritus of geology, Stanford University, who was in the vicinity, made a personal investigation of the effects of the earthquake and reported that "in general, good design and honest workmanship

stood." Badly designed concrete structures and steel buildings with brick veneer not tied in were destroyed.

It appears that Santa Barbara had no building code until last spring, when it adopted one, and this fact undoubtedly accounts for the poor construction which was formed in the seriously damaged buildings. Lessons learned from this and other earthquakes demonstrate that buildings can be designed at reasonable cost which will withstand the unusual stresses incident to shocks of this nature.

Inasmuch as the possibility of visitation by earthquake is practically undisputable, and as recent tremors have been felt in widely distributed sections of the country, East as well as West, it does not seem too much to expect that our modern buildings should be designed with this additional element of safety if merely from the standpoint of insurance alone.

Of the technical aspects of building construction civil engineers and architects are more vitally interested than are mechanical engineers. The important facts are that safe constructions are possible and at no excessive expense. All engineers are interested in insisting that only such constructions be allowed in modern cities.

A.S.M.E. to Meet at Altoona

The dates of October 5, 6, and 7, 1925, have been decided upon for the Regional Meeting of the A.S.M.E. to be held in Altoona, Pa. The technical program will deal with transportation problems and the excursions will include visits to the vast shops of the Pennsylvania Railroad System, as well as opportunities to enjoy the scenery of the lower Appalachians, which, at that time of the year, will be at the height of their glory.

The Council of the Society will hold its regular October meeting at Altoona to take up the budget for the fiscal year starting the first of October.

Asa Martines Mattice

An Appreciation by Walter M. McFarland

Counting from the beginning of his technical education, the professional life of Asa Martines Mattice covered just about half a century, and in every field of activity he enjoyed the confidence and respect of his colleagues in the highest degree. They appreciated his great professional knowledge, which was almost encyclopedic in its scope, his splendid skill as an engineer, and the remarkable quickness and accuracy of his mental processes.

His reputation began with his record at the Naval Academy, where he led his class. He entered the Academy with the second class of Cadet Engineers, for the two-year course. He was graduated in 1874 and was at sea on the *Brooklyn*, *Vandalia*, and *Trenton* until the summer of 1879.

In September, 1879, he was ordered to duty at the Naval Academy, where for the next three years he was an instructor in the Department of Steam Engineering. There, in association with Chief Engineer Kafer he developed a course in drafting and design which was the admiration of all who saw it, and the students in that course never ceased to speak of his high attainments as an instructor.

In 1882 he was ordered to sea on the *Miantonomoh* and the next year was transferred to the *Juniata*, which was the first foreign ship to bring succor after the eruption of Krakatoa, in the Sunda Straits, in which about half of an island was blown to atoms. She remained in the vicinity for months, taking soundings, directing shipping, making new charts, etc. in all of which Mattice had a leading part.

At the end of this cruise, he was ordered to duty in the Bureau of Steam Engineering, where he remained, except for a few months of detached duty, during the rest of his active service in the Navy. This was at the beginning of what is generally called in the Service the "New Navy." Chief Engineer, afterwards Admiral, Melville was appointed senior inspector at the Cramp Works, where several naval vessels were being built. Plans for some of these had been obtained abroad, but Melville, satisfied that he could produce equally good or better designs, obtained authorization from Secretary Whitney to prepare some designs. Chief Engineer Kafer, Passed Assistant Engineer Mattice and Assistant Engineers Cathcart, Bull, and Kaemmerling were ordered to assist him. There were no draftsmen to help these officers, who therefore had to do all the work of calculation and drafting. Professor Cathcart says: "Mattice was a living dynamo in his energy, speed, and capacity for work. In looking backward through the years to those torrid days of unremitting toil, the writer feels that Mattice, with his tireless energy, his swift work, and his vast fund of information, was a sort of *deus ex machina* in it all, supplementing Melville's genius and Kafer's surpassing skill." The design came into actuality as the machinery of the *San Francisco* and was the earliest of Melville's many successes in the design of machinery for our naval vessels.

Mattice was chief designer and in charge of the drafting room of the Bureau of Steam Engineering from August, 1887, to the end of December, 1888, when he went on leave for a year. He produced the design for the machinery of the original *Maine* (blown up in the harbor of Havana in 1898) and in connection therewith prepared the machinery specifications. These were the most elaborate and complete specifications made for an installation of marine machinery up to that time, and served for many years, with necessary modifications, as the model for specifications. Admiral Griffin, who was in the Bureau with Mattice, writes: "Mattice was responsible for the organization of the Bureau under Melville and this organization, with slight changes, has survived to the present time."

To those not fully cognizant of Naval history, this work may not seem remarkable, but engineering then in the Navy was at a rather low ebb. New vessels had not been authorized for years, and Naval officers had no intimate experience with the latest designs. That Mattice was able to do such splendid work was due, to quote Professor Cathcart, to the fact that "although his career was largely one of action, he was fundamentally a student, and, from boyhood till death came, his search for knowledge was unceasing. With this desire was joined a memory that very seldom failed, so that, as the years passed, his information on engineering subjects became encyclopedic." In other, words, although the machinery in service was not of the latest type, Mattice had been preparing himself, and, when the occasion offered, was ready to step in and do work of the highest class.

When he left the Bureau of Steam Engineering in 1890, he joined the staff of E. D. Leavitt of Boston, as principal assistant. W. H. Glocker, for many years superintendent of shops at Cramp's Shipbuilding Co., was chief inspector for Leavitt during Mattice's term there. He writes: "I considered him the best all-around engineer of my acquaintance. During his stay in the Cambridge office he had full charge of all designing, and he turned out plans for hoisting and pumping engines and mining machinery that remain unsurpassed for both quality and size." Mr. Mattice's close friend, Charles C. Tyler, writes: "It is my impression that Mattice's work, while with Leavitt, represented his highest achievements in engineering, and was the part in which he took the greatest pride."

After about ten years of splendid work with Leavitt, Mattice left him and was occupied in consulting work alone for about a year, when he joined the Westinghouse Electric Co. as chief engineer. At the end of three years, Mr. Westinghouse transferred him to the Westinghouse Machine Co. as chief engineer, but retained his connection with the Electric Company as consulting engineer.

While an absolute master of the theory of engineering, he was strongly practical, and one feature of his methods has always appealed to me very strongly. When a new design for a line of apparatus had been prepared in the drafting room, he always called in the foremen of the departments that would have to handle it, for inspection and criticism of the design. He even carried this so far that in some cases he had skeleton models made to be sure

that no parts conflicted. He could read a drawing as well as anybody, but experience had taught him that the time to avoid difficulties is in the preliminary stages and not after castings are made and work is begun. Dr. Hollis, in writing about Mattice, says: "I think he was a man of very remarkable intellectual qualities and really in many ways a great engineer. As a master of the technicalities of engineering, I doubt if he could have been surpassed in the United States."

Early in 1904, Mattice's old friend and classmate, B. H. Warren, became president of the Allis-Chalmers Co., and urged Mattice to go with him as chief engineer and in charge of manufacturing, which he did in April of 1904. Together, he and Charles C. Tyler, the general superintendent, planned the large extensions of the Allis-Chalmers works at Milwaukee.

After two years successful service with the Allis-Chalmers Co., Mattice resigned to become a partner in the newly organized firm of consulting engineers, Kafer, Warren and Mattice, with offices at 60 Wall St., New York. Mr. Kafer and Mr. Warren both died within a year after the organization of the firm, and their death so affected Mattice that he felt unable to continue the business alone. In 1907 he became works manager of the Walworth Manufacturing Co. of Boston, where he had full charge of the design and manufacture.

In 1911 he retired from active business, but a man of his energy could not be idle. To keep himself occupied and to lead an open-air life, he purchased a farm near Lockport, N. Y., where he raised fruit and chickens.

Meanwhile his old friend, Mr. Tyler, had become vice-president of the Remington Arms Co. When the great war began and the Allies applied to our establishments for munitions, the demand upon the Remington Co., of course, was enormously increased. Mr. Tyler besought Mattice to give up farming and come back into engineering to help him, because he knew that any work turned over to Mattice required no further supervision. Mattice consented and joined the Remington company in 1915, as advisory engineer. After the Armistice, he prepared most of the data required by the company for the settlement of its war contracts with the United States and other governments, and also in connection with its several government suits. His work was so carefully and thoroughly performed as to win the admiration, as well as the approbation, of the legal representatives of both parties to the suits.

During 1917 and 1918 Mattice was secretary of the Small Arms and Ammunition Manufacturers Production Committee, whose members comprised representatives from each of the manufacturers of these products, who had contracts with the United States Government. Mattice had been a member of The American Society of Mechanical Engineers since 1889 and was a Manager 1903–1906. He was also an original member of the Society of Naval Engineers and a member of the Engineers' Club of New York.

He was born in Buffalo, N. Y., August 1, 1853, the youngest of a large family, of whom only one sister survives. He had never married. He died at the Engineers' Club, New York, April 19, 1925.

Book Reviews and Library Notes

THE Library is a coöperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references to engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

EMPLOYEES' REPRESENTATION IN COAL MINES. A Study of the Industrial Representation Plan of the Colorado Fuel and Iron Company. By Ben M. Selekman and Mary Van Kleek, of the Department of Industrial Studies of the Russell Sage Foundation. Published by the Foundation in New York. Cloth, 5 × 7¾ in., 479 pp., $2.

Employees' Representation in Coal Mines

THERE can be no reasonable doubt that the authors have taken all possible pains to get at and to fairly and intelligently present the real facts connected with the history, development, and actual workings of this "Rockefeller Plan," as it has come to be known. Neither can the entire sincerity of Mr. Rockefeller, Jr., in his efforts to build up a plan to secure industrial peace be doubted. How far his efforts, supported by a portion of the employees and opposed by others, have been successful is here shown quite clearly. That there has been very great improvement in working conditions in the mines and also in the living conditions in the mining towns is very evident; as is also the fact that the miners and the miners' families keenly appreciate them. Few who read this book carefully and with a background of industrial experience can doubt that this plan, devised by or under the advice of the Hon. W. L. Mackenzie King whose work in calming the troubled waters of modern industrialism, first in Canada and later here, had previously attracted much favorable attention, has been devised with a very sincere desire by both Mr. Rockefeller and Mr. King to secure the best possible conditions obtainable under the circumstances.

But the peculiar circumstances were and are probably the greatest difficulty. It will be remembered that in 1913 the country was shocked by the massacre of a number of people living in a tent colony near one of the mines at Ludlow, Colorado. These people had been evicted from their homes on the property of the Colorado Coal and Iron Company as a result of a strike concerning wages, hours of work, enforcement of certain state mining laws, and union recognition. Among the killed were two women and thirteen children. This led to various "investigations" by governmental agencies and, more important perhaps, aroused the conscience of Mr. Rockefeller and evidently made him determined to see what could be done to avoid such conflicts and maintain peace in and about the mines.

In spite of the very great improvement that has taken place, however, it is made clear in this report that much remains to be done. The bitter memories of Ludlow; the fact that the plan provides for representation of the men by those of their own number, chosen from among fellow-employees who are fearful of discharge or of discrimination against them by petty bosses if they should undertake to do anything really worth while, all manifestly tend toward irritation and misunderstanding. Then there is the trouble which arises from the retention of minor executives who seem really to hate most of the men who work under them and miss no opportunity to decide against them when any issue arises. The plan provides explicitly for full liberty for any employee to join or not to join the miners' union and to participate in any of its activities in any lawful manner. But sometimes a petty boss undertakes to decide, all by himself, what is lawful and what is not, and also who are and who are not "trouble makers" and to arbitrarily act in accordance with his decision. Often these acts are repudiated when the records reach the higher stages of the

organization, but in too many instances such men seem to be retained in their positions to promote hatred; and, hatred, as we all know, is a poor lubricant for an industrial machine. In fact, there would appear to be much educational work to be done both among the minor bosses and the workers before the plan can work as well as, obviously, it is capable of working. Too often differences of opinion concerning payment for auxiliary or fortuitous work done by miners are decided arbitrarily by bosses who seem to think it their business to decide everything in what they conceive to be the interest of the company; their judgment of what is, or what is not in the interest of the company being often quite obviously wrong and eventually repudiated by higher authority. But many such cases naturally do not reach higher authorities; they are not taken up by the mens' representatives, for the reason previously mentioned. Other difficulties arise from the fact that the company usually owns the entire town in which the men and their families live. Though the company has made decided improvements in living conditions since the present "plan" went into effect, yet the combination of employer and landlord in the person of one man, or one company, does deprive men of some of the attributes and privileges of citizenship.

Strikes have occurred since the plan went into operation but they may be called "decent" strikes—no mine guards, no strike breakers, no violence, no shots fired.

The report again confirms the fact that workers are, after all, not to be judged as a class different from others, but that there are among them reasonable and fair men who will respond fully to fair treatment; also unreasonable and unfair men who will respond to nothing but force in some form—and all grades between these two extremes. In other words, they are just humans, like the rest of us, and no angels are procurable to work in coal mines, or anywhere else. We have to take these humans as they are and make the best of them; not the worst of them, as is too often done.

That is what the Colorado Coal and Iron Company seems to be trying to do in all sincerity and good faith; so far at least as its higher officers and principal owners are concerned. It is not yet perfect, but it has many excellent features, and anything is better than Ludlow or West Virginia.

In a footnote on page 396 it is said that similar plans are in operation in the International Harvester Company, The Standard Oil Company, the Pennsylvania Railroad Company, and several large meat-packing companies. Lack of more than a very general knowledge of the plans in use by these companies prevents me from making comparisons or drawing deductions from this statement, though I do believe they tend toward the promotion of industrial peace and coöperative working. It is interesting to note, however, that about the time the Pennsylvania Railroad adopted its plan the Baltimore and Ohio Railroad adopted a plan quite different in some of its most important features; and, while it is quite generally understood that the former plan does not by any means meet with universal approval (which is not saying it ought not to), the enthusiasm for the B. & O. plan seems to be well nigh if not quite unanimous by both management and employees. Perhaps the Russell Sage Foundation could do no greater service than to look into the differences between these two plans in the same industry and in substantially the same territory and give us a comparison of their actual workings. If undertaken we may be pretty sure the work will be conscientiously and competently performed.

Fred J. Miller.[1]

Books Received in the Library

A Course of Metallurgy for Engineers. By F. C. Thompson. Witherby, Lond., 1925. Cloth, 6 × 9 in., 240 pp., illus., diagrams, tables. 27s, 6d net.

Discusses the composition and structure of the metals used in engineering, the defects in ingots, heat treatment, the changes induced by hot or cold working, case-hardening, etc. The text is confined to iron, steel, brass, bronze, aluminium alloys, and bearing metals. The text is compressed, yet readable, and is intended for the user of forgings and castings rather than for the maker.

[1] Member, Public Service Commission of Pennsylvania, Harrisburg, Pa. Past-President, A.S.M.E.

Dynamik, vol. 1; Dynamik des Einzelkörpers. By Wilhelm Müller. Walter de Gruyter & Co., Berlin and Leipzig, 1925. Cloth, 4 × 6 in., 160 pp., diagrams. 1.25 R.M. (.30).

Although so small, a very terse style has enabled the author to cover the essentials of the subject.

Effective Regulation of Public Utilities. By John Bauer. Macmillan, New York, 1925. Cloth, 6 × 8 in., 381 pp., $2.50.

The purpose of this book is to consider critically the existing policies and methods by which the regulation of public utilities has been attempted; to show the inadequacy and deficiency of the existing machinery; and to suggest constructive measures for a realization of the fundamental purposes of regulation. It aims to give the accountants, engineers, and lawyers interested in the subject a clearer, more vivid understanding of what is involved in regulation and what is needed to make it effective. The author has been engaged in regulation for fifteen years and is public-utility consultant to the Corporation Counsel of the City of New York.

Die Elastischen Platten. By A. Nádai. Julius Springer, Berlin, 1925. Boards, 6 × 9 in., 326 pp., illus., diagrams, tables. 24 G.M.

An extended mathematical study of the changes of form of and the internal stresses in slabs and plates. The author aims to acquaint mechanical and structural engineers with the best methods for determining the effects of loads upon slabs and to provide a summary of our knowledge at the present time.

Die Gewinde. By G. Berndt. Julius Springer, Berlin, 1925. Boards, 6 × 9 in., 657 pp., illus., diagrams, tables. 36 G.M.

An exhaustive account of the history of screw threads, of the methods of manufacture, and of testing. The first section gives a thorough account of the evolution of the different thread systems from their origins to the present time. The second section treats in detail of methods of measurement and of instruments for that purpose. Section three discusses tolerances and testing. The volume covers its field very fully and critically and is a valuable, authoritative summary of the subject. American and European practice is covered, and a good bibliography is given.

Heat Engines, Steam, Gas, Steam Turbines and their Auxiliaries. By John R. Allen and Joseph A. Bursley. Third edition. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 422 pp., illus., diagrams, tables. $4.00.

An elementary treatise based on the course given at the University of Michigan. Only those engines are considered which are in common use, and the use of the calculus and higher mathematics has been avoided. The forms of heat engines discussed included the steam engine with its boiler plant, the gas engine with its producer, oil engines, and steam turbines. This edition has been rewritten to a large extent.

Hütte, Taschenbuch für Betriebsingenieure. Published by Akad. Verein Hütte und A. Stauch. Second edition, revised and enlarged. Cloth, 6 × 7 in., 1325 pp., diagrams, tables, 19.50 G.M.

This latest edition to the well-known series of handbooks is prepared for mechanical engineers and works managers, as a concise guide to shop organization and management. The twenty-seven sections treat of the properties, testing, and strength of materials, power transmission, measuring instruments, standardization, factory planning, conveyors, accident prevention and shop hygiene, shop organization, time study, inspection, training of workers, social and political relations, foundry practice, welding, brazing, forging and stamping, hardening, laying out castings and forgings, cutting tools, compressed-air tools, electric tools, jigs, machine tools, woodworking tools, and balancing. The book is the work of a group of specialists. Its popularity is indicated by the exhaustion of the first edition within eight weeks after publication.

Die Leuchtgasindustrie. By Arthur Fürth. de Gruyter, Berlin, 1925. Cloth, 4 × 6 in., 132 pp., illus., diagrams, 1.25 R.M.

Describes briefly the modern illuminating-gas plant and the methods of manufacture and utilization. Chapters are devoted also to gas analysis, by-products, and the economic situation of the industry.

THE ENGINEERING INDEX

Registered United States, Great Britain and Canada

LAST-MINUTE ADDITIONS; MAIN BODY ON PAGE 151-EI, ADVERTISING SECTION

Exigencies of publication make it necessary to put the main body of The Engineering Index into type considerably in advance of the date of issue of "Mechanical Engineering." To bring this service more nearly up to date is the purpose of this supplementary page of items covering the more important articles appearing in journals received up to the third day prior to going to press.

ACCIDENT PREVENTION

Steel and Wire Plants. Putting a Premium on Safety, J. Nelson. Iron Age, vol. 116, no. 5, July 30, 1925, pp. 269–271, 1 fig. Particulars of accident-prevention system of Worcester, Mass., works of Am. Steel & Wire Co.; each year sees fewer accidents.

AIR COMPRESSORS

Explosions in. Avoiding Compressor Explosions, T. O. Organ. Power, vol. 62, no. 3, July 21, 1925, p. 87. Effect of too much oil; thin oil with high flash point should be used.

AIRSHIPS

Water Recovery. The Uses of Water Recovery, H. F. Parker. Aviation, vol. 19, no. 1, July 6, 1925, pp. 6–7. Water recovery is now in use as regular equipment on airships *Shenandoah* and *Los Angeles*, and it has been conclusively demonstrated that amount of water can be recovered from exhaust gases from engines, in flight, which more than equals weight of fuel consumed in producing those exhaust gases; advantages in maneuvering.

ALUMINUM

Castings. Cast Aluminum Washer Tube, P. Dwyer. Foundry, vol. 53, no. 14, July 15, 1925, pp. 556–559 and 563, 8 figs. Through installation of mechanical equipment for reconditioning and handling sand, molds, and castings, intensive production is secured; practice at new aluminum foundry of Maytag Co., Newton, Ia.

AUTOMOBILE ENGINES

Journal Bearings, Oil Flow in. Oil-Flow in Complete Journal Bearings, D. P. Barnard, 4th. Soc. Automotive Engrs.—Jl., vol. 17, no. 2, Aug. 1925, pp. 205–209, 9 figs. Develops general laws governing rate of flow of oil through complete journal bearings; these laws are based on assumption that axial flow obeys Poiseuille's law and is, therefore, a function of bearing load; general relation of rubbing speed to heat generation and oil flow is discussed for purpose of indicating a possible solution of certain high-speed-bearing problems.

AUTOMOBILES

Transmissions. Friction Transmission for Automobiles, C. A. Trask. Soc. Automotive Engrs.—Jl., vol. 17, no. 2, Aug. 1925, pp. 210–211, 4 figs. Notes on adaptability to light cars, and advantages.

CASTINGS

Aluminum, X-Ray Examination of. Probe Aluminum Castings with the X-Ray, R. J. Anderson. Foundry, vol. 53, no. 15, Aug. 1, 1925, pp. 606–608, 11 figs. Advantages of X-ray examination in production and use of X-rays; how a radiograph is made, apparatus required.

CENTRAL STATIONS

England. The Ribble Power Station of Preston Corporation. Engineering, vol. 120, no. 3107, July 17, 1925, pp. 73–76, 12 figs. partly on supp. plates. Description of station, dealing with history, coal- and ash-handling equipment, arrangement of boiler house, sub-stations, and equipment; contains two turbo-generators each having a continuous normal rating of 12,500 kw. at 0.9 power factor and delivering three-phase current at 50 cycles and 6600 volts. See also Engineer, vol. 140, no. 3630, July 24, 1925, p. 96.

CHUCKING MACHINES

Automatic. Spindle Speeds of 2000 R.P.M. Possible on New Chucking Machine. Automotive Industries, vol. 53, no. 5, July 30, 1925, pp. 185–186, 3 figs. Describes new 12-A New Britain automatic chucking machine; chucks opened automatically for most classes of work and operator is required only to close them by use of air control valve after insertion of rough piece.

COKE OVENS

Coke Pushing and Coal Leveling Machines. Coke-Pushing and Coal-Levelling Machine. Engineering, vol. 120, no. 3107, July 17, 1925, pp. 66–68, 2 figs. Describes machine designed by Wellman Smith Owen Engineering Corp., Ltd., England, for performing the three operations of door handling, coal levelling, and coke extraction; machine runs on rails laid parallel with face of coke ovens, and carries three independent mechanisms for performing operations specified.

CONVEYORS

Foundries. Conveyor System Cuts Costs. Iron Age, vol. 116, no. 6, Aug. 6, 1925, pp. 336–338, 3 figs. Saving of 50 per cent in floor space effected by Interstate Foundries, Inc., Cleveland, O., through installation of continuous power-driven conveyor equipment for handling molds, flasks, bottom boards, castings, and sand. Method of handling work on conveyors.

DIES

Sub-Press Compound. Standardized Sub-Press Compound Dies, W. E. Irish. Am. Mach., vol. 63, no. 5, July 30, 1925, pp. 177–181, 6 figs. Construction, operation and standardization of sub-press dies for small stampings in large quantities.

FORGING

Automobile-Engine Connecting Rods. Forging and Machining Connecting Rods, F. H. Colvin. Am. Mach., vol. 63, no. 4, July 23, 1925, pp. 155–157, 7 figs. Use of forging machines and special dies to secure rods that require but little machining; special fixtures and machining also used.

FOUNDING

Condenser Shells. Makes Large Condenser Shell, H. A. Hart. Foundry, vol. 53, no. 15, Aug. 1, 1925, pp. 601–650, 12 figs. Methods employed in jobbing foundry of River Rouge plant of Ford Co. in making condenser shell in connection with production of castings for specially designed steam turbo-generators for main power house at River Rouge plant.

GRINDING MACHINES

Internal. High Production Capacity a Feature of New Internal Grinder, P. M. Heldt. Automotive Industries, vol. 53, no. 3, July 16, 1925, pp. 86–87, 3 figs. Describes new internal grinding machine for rapid production work in which many of movements that formerly had to be effected by operator are accomplished automatically, developed by Heald Machine Co. of Worcester, Mass.; sizing of parts is done by means of a direct-reading indicator.

Roll. Builds Large Roll Grinder. Iron Age, vol. 116, no. 5, July 30, 1925, pp. 279–280, 4 figs. Describes grinding machine built for Mesta Machine Co., West Homestead, Pa., by Landis Tool Co., Waynesboro, Pa.; has 50-in. swing, 24 ft. between centers, and incorporates new features; takes rolls weighing up to 28 tons.

HYDRAULIC TURBINES

Corrosion. The Problem of Corrosion of Turbine Rotors (Zur Frage der Anfressungen von Turbinenlaufrädern), Feifel. Zeit. des Vereines deutscher Ingenieure, vol. 69, no. 24, June 13, 1925, pp. 815–818, 6 figs. Rotors of large turbine installation operating since 1918 showed after short operating period serious corrosion; based on experiences gathered in Germany and abroad, and according to rules of design developed from these experiences, operating conditions of this plant must be regarded as altogether unfavorable; account of successful replacement of endangered rotors; new rotors may be regarded as free from corrosion with promise of normal life.

HYDROELECTRIC DEVELOPMENTS

San Francisco, Cal. The Hetch Hetchy Power Development, M. M. O'Shaughnessy. Elec. Wld., vol. 86, no. 5, Aug. 1, 1925, pp. 206–211, 10 figs. Describes Moccasin power plant, first power unit on Hetch Hetchy municipal water and power project, which is now delivering electrical energy to San Francisco; has four units with a total rated capacity of 80,000 kva., and provision has been made for addition for two more.

INDUSTRIAL ORGANIZATION

Engineering Department. Some Comments on Engineering Department Organization, P. H. Bryant. Am. Mach., vol. 63, no. 6, Aug. 6, 1925, pp. 233–235. Outline of requirements for organizing a small engineering department.

INTERNAL-COMBUSTION ENGINES

Air Cleaners. Air Cleaner Combines Principles of Screen and Oily Surfaces. Automotive Industries, vol. 53, no. 3, July 16, 1925, p. 92, 2 figs. Particulars of new air cleaner for internal combustion engines and similar purposes placed on market by Air-Maze Corp. of New York City; claimed to offer negligible resistance to air flow and to eliminate all dust.

LOCOMOTIVE BOILERS

Pitting. Causes and Prevention of Pitting in Locomotive Boilers, C. H. Koyl. Ry. Mech. Engr., vol. 99, no. 8, Aug. 1925, pp. 517–518. Discusses pitting caused by acids, pitting caused by electric action, causes and prevention of electric pitting, and eliminating oxygen in feedwater.

LOCOMOTIVES

4-10-2. Three-Cylinder 4-10-2 Locomotives for the S. P. and U. P. Ry. Age, vol. 79, no. 6, Aug. 8, 1925, pp. 268–271, 3 figs. Particulars of locomotives which are development from 2-10-2 type in which two-wheel leading truck has been replaced by a four-wheel truck; tractive force of Southern Pacific with booster is 95,700 lb., and that of Union Pacific with no booster is 78,000 lb.

Oil-Electric. Oil-Electric Locomotive Performance. Ry Age, vol. 79, no. 5, Aug 1, 1925, pp. 231–233, 5 figs. Performance data in different services and under different conditions of 60-ton oil-electric locomotive built jointly by Am. Locomotive Co., Ingersoll-Rand Co., and Gen. Elec. Co.

Transmission Gears. The Schneider Hydro-Mechanical Transmission Gear. Engineer, vol. 140, no. 3630, July 24, 1925, pp. 86–89, 11 figs. partly on p. 90. Describes new type of hydraulic transmission gear designed by H. Schneider, chief engineer of Swiss Locomotive & Machine Wks., Winterthur, for large powers; a direct mechanical connection between primary and secondary elements; operation; results of tests.

MACHINERY

Cardboard-Box-Making Machines. A Cardboard Box Machine. Engineer, vol. 140, no. 3630, July 24, 1925, p. 94, 4 figs. Describes a double piecing-on or ending-on machine, made by M. C. Ritchie, Ltd., London, England; used for first operation in making cardboard boxes, such as those used for stationery, after blanks have been cut out, and fixes ends on to other piece of card, which forms bottom and sides, by means of strips of gummed paper.

METALS

Physical Properties. The Significance of Common Physical Properties of Structural Materials, F. H. Moore. Soc. Automotive Engrs.—Jl., vol. 17, no. 2, Aug. 1925, pp. 175–180 and (discussion) 180–182, 3 figs. Author summarizes his views as a testing engineer regarding significance of results of some of the common physical tests that are applied to metals; limitations of commercial material-testing processes.

MACHINE TOOLS

Lapping Machines. Lapping Machine for Repetition Cylindrical and Flat Work. Engineering, vol. 120, no. 3107, July 17, 1925, pp. 90–91, 8 figs. Describes machine built by Bethal, Player & Co., Ltd., London, England; made in two forms, one designed for lapping cylindrical pins or tubes, and other for dealing with flat parallel faces.

Radial Drilling Machine. 6-Ft. Radial Drilling Machine. Engineering, vol. 120, no. 3108, July 24, 1925, pp. 103 and 108, 6 figs. Describes radial drilling machine designed and constructed by George Swift & Sons, Ltd., of Claremont Iron works, Halifax, for use either as a single unit or as part of a battery.

Threading Machines. Double-End Automatics Designed for Threading and Chamfering Operations, W. L. Carver. Automotive Industries, vol. 53, no. 4, July 23, 1925, p. 139, 2 figs. Notes on two automatic machines built by Grant Mfg. & Machine Co. of Bridgeport, Conn., a double-end machine for threading both ends of studs simultaneously, and a double-end machine which is adapted to facing, chamfering and similar operations on bushings, pins, etc.

MILLING CUTTERS

Cutting Speed and Pressure. Cutting Speed and Cutting Pressure in Milling (Ueber Schnittgeschwindigkeit und Schnittdruck beim Fräsen), G. Engel. Zeit. des Vereines deutscher Ingenieure, vol. 69, no. 24, June 13, 1925, pp. 819–822, 8 figs. Based on Friedrich formula for favorable cutting speed with turning, suitable equations are developed for milling with and without chip-breaking grooves, and also equations for cutting pressure; comparison with former cutting-pressure tests demonstrates correctness of these formulas.

OIL

Viscosity Tests, Standardization of. Plan Worked Out for Standardization of Oil Viscosity Tests. Automotive Industries, vol. 53, no. 4, July 23, 1925, pp. 136–138. Recommendations of committee of Am. Petroleum Inst. for eliminating discrepancies in results obtained by different observers with same oil and instrument.

SAND, MOLDING

Testing. The Present Status of the Laboratory Investigation of Sands, H. Ries. Foundry Trade Jl., vol. 116, nos. 460 and 461, June 11 and 18, 1925, pp. 495–498 and 523–525, 6 figs.; also Foundry, vol. 53, nos. 13 and 14, July 1 and 15, 1925, pp. 531–534 and 546, and 576–579, 6 figs.; and Metal Industry (Lond.), vol. 26, no. 24, June 12, 1925, pp. 579–582, 1 fig. Deals chiefly with what has been done in United States, but makes references to and comparisons with foreign tests; fineness test and figure; grading sands; bonding strength; bar test; compression and tensile-strength tests; density of sand; optimum water content; moisture determination; refractoriness and life; longevity of sands; chemical analyses; dry-sand testing. American exchange paper to conference of Inst. Brit. Foundrymen.

STEAM

Research. Steam Research, I. V. Robinson. Power, vol. 62, no. 4, July 28, 1925, pp. 134–136, 3 figs. Outlines work which Prof. Callendar is carrying on in England. Three types of apparatus are used, all working on steady-flow principle, comprising a throttling calorimeter to determine total heat by comparison, an electric calorimeter for direct measurement of specific heat at constant pressure and a condenser for direct measurement of total heat, all ingeniously designed not only to reduce heat losses to a minimum, but also to permit accurate determination of heat-loss corrections.

WIRE MILLS

Gas as Fuel in. Wire Mill Uses City Gas for Fuel, J. M. Layng. Iron Age, vol. 116, no. 5, July 30, 1925, pp. 275–278, 4 figs. All furnaces at plant of Stewart Hartshorn Co., Newark, N. J., such as open annealing, pot annealing, baking, patenting and tempering, have been redesigned for use of city gas; higher output and uniform quality at lesser cost.